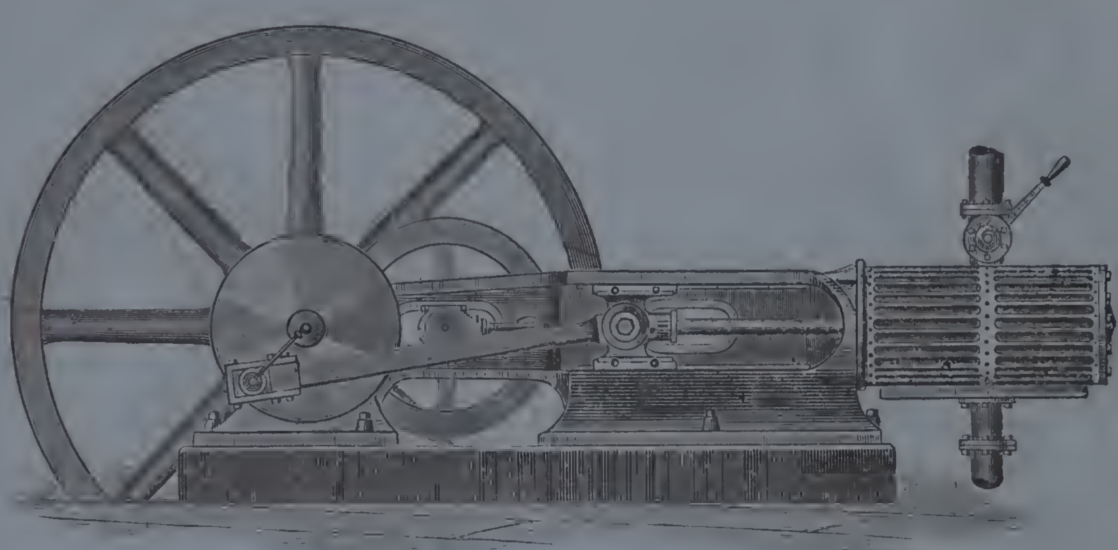
ILLUSTRATED CATALOGUE

—OF—

THE CUMMER ENGINE CO.



CLEVELAND, OHIO.

HILL, CLARKE & CO., EASTERN AGENTS,
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ILLUSTRATED CATALOGUE

OF THE

CUMMER ENGINE CO.

MANUFACTURERS OF

Automatic Steam Engines,

SPECIALLY ADAPTED FOR

ELECTRIC LIGHTING,

FLOURING, COTTON AND WOOLEN MILLS,

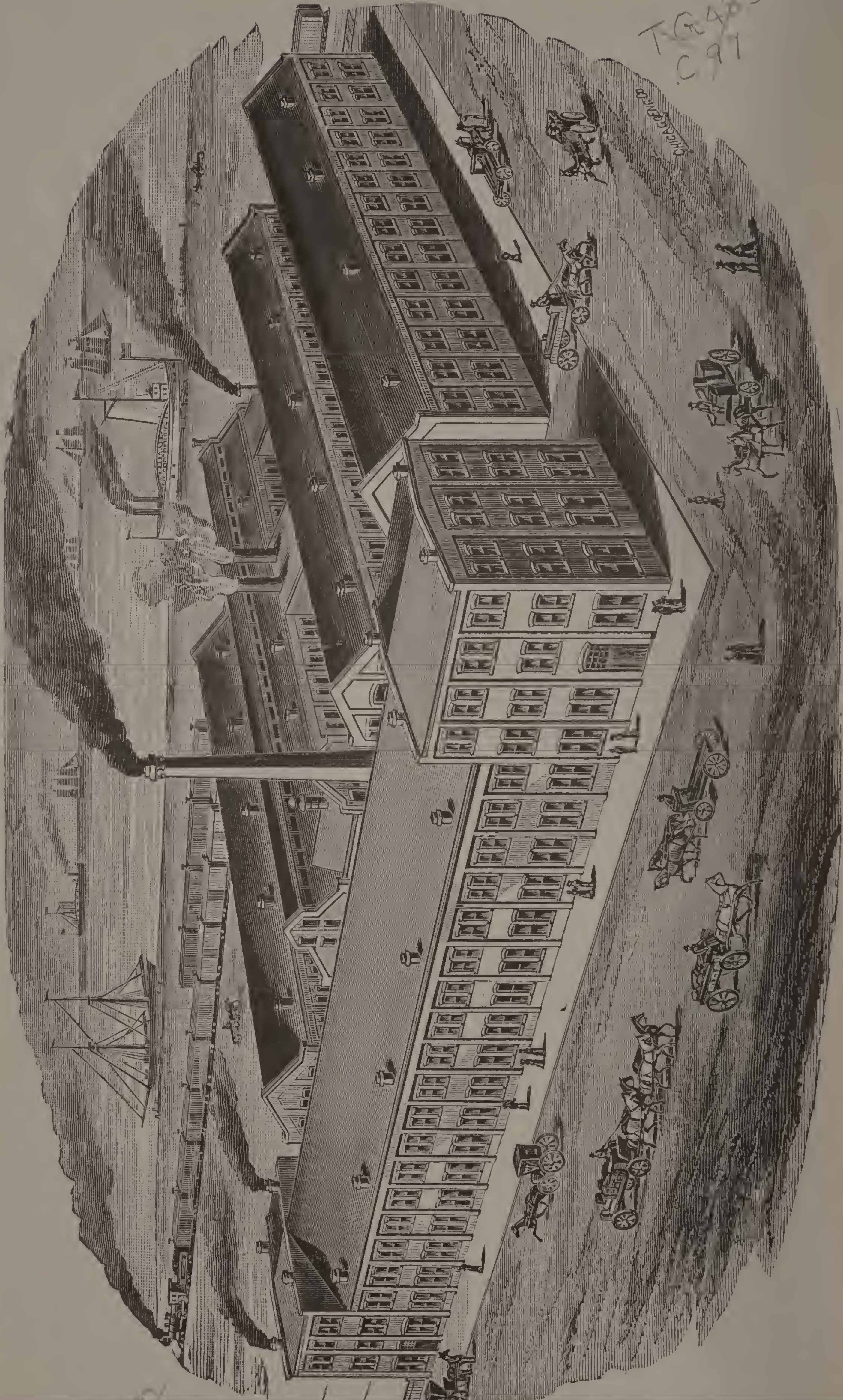
AND ALL MANUFACTORIES REQUIRING

Unusual Steadiness of Motion,

OFFICE AND WORKS:

LAKE STREET, NEAR KIRTLAND,

CLEVELAND, O.


CHICAGO ENG. CO.

INTRODUCTION

The works of this Company are entirely new, having been erected during the year 1882. They are eligibly located, possessing unusual shipping facilities, as well as being admirably adapted for our business.

The accompanying engravings represent the entire plant as designed for our use.

The tools have been built for us with especial reference to doing good work rapidly, many of them were designed and constructed with a view to the production of better work than could be done with ordinary machine tools. In addition to the larger tools especial attention has been given the smaller tools, jigs and gauges, in order that our work may be made strictly interchangeable; a feature which we have secured at a heavy outlay. The advantages to the purchaser by reason of this special adaptation of tools toward the production of the very best work will be appreciated by all who may feel inclined to inspect our method of doing work.

SIMPLICITY OF MECHANISM.

It has been our aim to make all our engines as simple as possible. Wherever we could safely do so, we have reduced to the fewest possible number of parts, the train of mechanism designed for operating the cut-off.

In no case, however, have we neglected to put in all those parts which are not only essential, but advantageous to the use of steam.

ACCESSIBILITY OF PARTS.

We think this feature in our engine, one which will meet with the hearty approval of engine-men everywhere. It has been our constant study to avoid anything which would make our engine difficult to adjust, or to repair in case of accident. We have endeavored as far as possible to have everything movable in plain sight; to have everything requiring adjustment to be within easy reach with ordinary tools.

LUBRICATION.

This very important matter has been well considered by us in the designs for all the several wearing surfaces in our engines. We understand full well the value of thorough lubrication, and no efforts have

been spared on our part to make the distribution of oil as complete as possible. Improved devices for oiling the valves and piston of our engines will always attract our attention and will be found to accompany each engine made and shipped by us.

ADJUSTMENT FOR WEAR.

The wearing surfaces in all our engines are not only ample for the service required, but are provided with means of adjustment in case of wear. Reference is had in the descriptive matter in this catalogue to some of the methods by which the adjustment of parts is secured ; an examination of some of the drawings, such as, the main bearing, cross-head, or connecting-rod, will show a well considered attention to this important matter of detail.

EXCELLENCE OF WORKMANSHIP.

This Company has availed itself of everything in the way of the adaptation of special tools to the securing of perfect surfaces, whether flat or cylindrical, that are needed in the construction of steam engines. Our system of gauges, jigs, and measuring instruments is quite complete, and so far as they apply to our particular work, leave nothing to be desired for the attainment of true surfaces. All wearing surfaces are scraped to a perfect fit. The valves are scraped independently to accurate surface plates, and the valve seats are made to correspond to the accuracy of the valves. We do not permit the use of emery in our works in fitting flat surfaces.

All holes are carefully reamed by hand which require accurate fitting ; the holes being tested by solid plugs, and the turned work to rings, to insure the utmost nicety of adjustment of parts to each other, the proper allowance being made in all cases requiring lubrication. Bolts, nuts and studs are cut with threads corresponding to the Franklin Institute standard, which has been adopted by the U. S. Government, and by the leading engineering establishments in this country. All wrought iron work subject to wear is case hardened.

ELEGANCE OF FORM.

It has been our aim to produce not only a strong and well proporioned engine, but at the same time to give the engine a graceful outline, preserving as far as possible those curves which are always pleasing to the eye rather than the rigid outlines which are the result of connecting

straight lines. We are sure our designs will compare favorably with any yet brought out; we have endeavored so far as circumstances will admit in making the outlines pleasing to the eye without in any degree sacrificing utility or good proportion.

MINOR DETAILS.

We have endeavored above all things to have the details of our engines perfect in every part, and well adapted for the uses intended. We have aimed higher than to merely present a good appearing engine as a whole, but have given the most minute detail as much consideration and thought, wherever needed, as we have given the larger or more conspicuous pieces. Many of our costly special machines have been designed and built by us for the express purpose of making the minor parts of our engines better than would be practicable by ordinary methods. A careful inspection of our engines will satisfy anyone that no detail however slight, is out of harmony with the general design.

LARGE WEARING SURFACES.

No subject in connection with the design of the details of our engine has received more attention than the wearing surfaces. It is of the utmost importance that they be as large as the circumstances of the case will admit. Especial attention has also been given the kind of material entering into the several parts which are brought in contact; the choice of metals being made upon the sole condition of efficiency and durability. Our proportions will be found to be very liberal, having been calculated first, for strength, and then for thorough lubrication in such portions of the engine as are subject to high pressures. These two problems should always be considered at the same time, for the reason that, it is not enough that the parts be strong enough to do the work, there must also be at all times perfect lubrication if the parts are to last any considerable length of time. For this reason we advocate large surfaces.

We invite attention also to the superior workmanship which we employ in the fitting of all wearing surfaces. Cylindrical surfaces are carefully tested, both for perfect roundness and for perfect parallelism; these having been secured, the boxes or bearings, are then scraped to a perfect fit. All flat surfaces are scraped to surface plates independently of each other, thus insuring the best possible surface for wear, as well as getting the surfaces out of wind.

The main bearing is very large and has means of adjustment which are described in the section pertaining especially to it. The wearing surface is such that with the very heaviest wheel which any given engine

will ever be expected to use, ample lubrication is afforded by the smaller pressure exerted by reason of the increased surface, consequent upon a long bearing of larger diameter than is usual or customary among engine builders.

SELECTION OF MATERIALS.

Probably no city in this country is more favorably located with reference to a proper selection of materials for engine building than the one in which our works are erected. We have a choice of the very best pig iron, wrought iron and steel, at prices which are as favorable as any other point in the United States.

In the selection of materials, as in everything else, we will have only the best. The pig iron for our cylinders is selected with special reference to hardness, closeness of grain, and great strength. The iron for the other castings are selected with reference to strength and rigidity rather than for hardness. Steel is used wherever its qualities recommend it over the employment of wrought iron. For crank pins subject to great stress, as well as hard wear, we have a special hammered crucible cast steel made for us, with just sufficient carbon in it to make a firm homogenious metal. Our valve rods are made of the same material, we giving it the preference at the increased cost over open hearth, and Bessemer rolled steels, because of its greater homogeneity and the entire absence of seams which occur in all rolled steels. Wherever gun metal is used the mixtures are those which give us the greatest hardness and toughness for the place intended. We use only new copper and tin in our mixtures for gun metal, varying the proportions according to the hardness required. The anti-friction metal used by us is made according to our own formulæ, from new metals, and is the very best metal we know of for the places in which we use it.

The iron and steel forgings are made from the best stock, and worked under very heavy hammers, thus insuring thorough working and welding to the center.

CLEARANCE.

The clearance in our engines has been brought down to the lowest possible limit. The shorter the stroke of the engine the greater will be the percentage of clearance as compared with engines of long stroke. The connecting rods on our engines are made so that shortening of the rod between centers does not occur, except, so far as one pair of brasses may wear more than those at the other end of the rod, this

enables us to allow less clearance between each end of the cylinder heads and piston. The steam and exhaust passages are made as small as is consistent with the actual requirements of the engine.

In setting the valves we are particular to see that on the low crank angles compression begins at that portion of the stroke as will give between the piston and cylinder head a pressure corresponding nearly or quite to that in the boiler. The effect of this cushioning is to raise the pressure of the exhaust steam at the end of the stroke, and thereby raise the temperature up to that of the steam to be admitted for the return stroke. As our steam and exhaust valves can be changed from the outside of the steam chest these changes may be made at any time when the indicator diagrams show it to be necessary.

STEAM PRESSURE.

We recommend a moderately high steam pressure, say from 80 to 90 pounds per square inch in the boiler. The advantages of high pressure steam are too well known to need any explanation here. We do not advocate extreme pressures, such as 150 pounds per square inch and upwards, because of the greater strength required in the boilers, fittings, etc., which add increased cost without a corresponding increase of efficiency and economy. Our engines are made capable of withstanding any practicable steam pressure, but common experience has demonstrated that an excellent economy can be had at the pressures named at the beginning of this paragraph. It sometimes happens, especially in the purchase of an engine for a new establishment, that the engine is much too large for the work; in this case we recommend a modified pressure, based on the indicator diagram, taken with an average load on the engine the pressure should be such that the lowest portion of the expansion line should be above the atmospheric line at the moment of the opening of the exhaust valve; and the steam pressure should be lowered until such a line can be traced by the indicator. As the load is increased so also should the steam pressure be increased, until the 80 or 90 pounds are had, which, in cutting off at 1-5 to 1-4 of the stroke, will, with our engine, yield high economy and with very satisfactory results. Our arrangement of valves is such that a full boiler pressure can be had at the beginning of the stroke.

AUTOMATIC CUT-OFF ENGINES.

Automatic Cut-off Engines are those in which steam of full boiler pressure is admitted to the cylinder and allowed to follow as far as may be necessary for the work required of the engine, and then cut off by some train of mechanism included in the engine itself, the steam being

allowed to follow at full boiler pressure for a portion of the stroke, and for the remaining portion is worked expansively; the regulation of speed for varying loads is obtained by varying the point of cut-off, and therefore the mean effective pressure. Certain economical limits of cut-off are established for the engine when designed, and the governor controls the admission valve in such a way as to cut off according to the load and maintain regular speed. To properly compare this class of engine with others, it may be well to say a few words about steam expansion, although the benefits obtained by using high pressure steam with considerable expansion are now well understood and it will not be necessary here to go into any extended discussion of the matter.

The most economical point of cut-off, however, is still an open question, and is to be settled rather by practical consideration than by theory. There can be no doubt that, beyond a certain point, the losses from internal condensation and other sources, together with the increased cost of the engine, become so great that it does not pay to further increase the rate of expansion. There is for each pressure a certain economical limit, which depends upon all these considerations, and each maker must determine for himself what cut-off to adopt. In our own practice, with steam at 90 lbs., we consider that a cut-off at $\frac{1}{5}$ or $\frac{1}{4}$ gives excellent economical results, and the ratings of our engines are based upon these figures. We believe $\frac{1}{4}$ to be rather the most economical, all things considered. The economy of expansion being conceded, it is desirable to know what method secures the best result. Leaving out of consideration compound engines, expansion may be obtained by using either a fixed or a variable cut-off. In the former case steam is cut off at a certain definite fraction of the stroke, which point having once been determined upon, remains fixed. These engines are regulated by a governor so connected with a throttle-valve as to give, according to the load, more or less steam opening, and consequently more or less initial steam pressure. The lowering of the initial pressure by causing the steam to flow through a contracted opening, is known as "wire drawing" and is always a direct loss, for instead of having full boiler pressure in the cylinder we can only count upon about four-fifths of the pressure and this much is seldom obtained, owning to the faulty designing of most builders of this class of engines. Governing by varying the initial pressure is thus a wasteful mode and not only this, but the engines cannot maintain steady speed which, besides being a serious disadvantage in many cases requiring uniform speed, is a still further waste of steam. A much closer regulation and far better economy is obtained by varying the point of cut-off, and consequently the mean effective pressure, instead of varying the initial pressure. In this way there is no loss by wire drawing and we use only that quantity of steam which the load requires. Such a regulation is accomplished by

automatic engines, where the governor takes full control of the steam supply and determines the point of cut-off, making it occur earlier or later as the load changes and admitting only the necessary quantity of steam. Automatic engines have a further advantage in this, they can use steam of considerably higher pressure than the throttling engines, because the latter will not regulate properly with a high pressure, nor can they cut off earlier than the one fixed point, while automatic engines can use high pressure steam with a range of expansion suited to the work. But besides economy in steam which is a matter of great importance to most manufacturers, it is often not less important, and sometimes indispensible, to have a very steady speed. Automatic engines can govern much more closely than any other kind of engine, in consequence of the governor having direct control of the steam admission and always adapting the mean effective pressure to the work to be done at each instant, and they are invariably used in situations where uniform speed is required.

COMPARISON OF AUTOMATIC ENGINES.

The various makes of automatic engines may, in general, be classed under either one of two types known as the releasing and positive, so called from the method by which the cut-off valve is operated; each of these types of engines have a variable cut-off controlled by the governor, though the governors differ in detail from the nature of their work. In the releasing type of automatic engine, such as the Corliss, and others, the governor employed is of the ordinary fly-ball form; the cut-off valve is not directly connected with the eccentric, but takes its motion from it through the intervention of tappets. The length of time these tappets shall remain in contact, and therefore the length of time the valve shall remain open is determined by the governor; which, as the balls rise and fall, releases the valve at an earlier or later point of the stroke. As soon as the valve is released it closes instantly from the action of a spring or other suitable device. For the return stroke the tappets are brought into contact and remain so until released through the action of the mechanism connected with the governor. This hooking on and releasing must take place at every stroke, and the striking of the tappets becomes such an objectionable feature that engines of this type have to confine themselves, for successful working, to comparatively slow speeds. The valve in this kind of engine closing the moment it is released, gives a sharp corner to the diagram, and the valve also is well open at the beginning of each stroke. The governor itself is simple, but there is required always a more or less complicated system of tappets, cams and springs to connect and disconnect the valve with the eccentric rod which gives it motion; and these, besides being complicated, are always liable to get out of order.

In that class of engines in which the valve has a positive motion, the governor is at all times connected with the valve, and the cut-off is varied by shifting the eccentric around the shaft so as to change its angularity, and point of closure of the valve. The various governors used in this kind of engine have points of resemblance, and they consist essentially of weights revolving in a vertical plane whose centrifugal force is intended to be balanced at all speeds by the tension of a spring, so that the governor weights can move out or in, and by suitable mechanism be made to operate the cut-off valve. Quite a number of governors of this class have been brought out during the past ten years with more or less improvement, but still retaining in some form or another the objectionable features of the original, which consisted in employing a spiral or elliptic spring to counteract and have under constant control the centrifugal effect produced or exerted by the revolving weights. As neither the centrifugal force or the tension of the spring remains constant at any two changes of position of either, it will be seen that it is an extremely difficult task to govern closely when one variable force is held in check by another variable force.

An immense advance was made in the construction of governors of this particular class when the revolving weights were made to lift a dead weight; in other words, when the centrifugal force of the flying weights was made to lift a weight which represented a force as constant as gravity. The Cummer governor is of this description, but the mode of its balancing the centrifugal force and the attendant advantage as well as other advantages arising from the arrangement and construction of the several parts are fully set forth under the head of THE GOVERNOR. The positive motion valve cannot have so quick a closure as a release valve, and theoretically this is a disadvantage, but practically it amounts to little, as may be seen by examining the indicator diagrams taken from our engine. At the point of cut-off, the corner is slightly rounded; this is due to the valve being gradually instead of instantly closed and the steam is wire-drawn to this extent. In engines of this class, however, there is no limit in either direction to the speed at which they can be run, while the drop cut-off variety of engines, in consequence of having such a complicated system of tappets, cams and springs, can only hook on to the valve and release it a limited number of times per minute. Engines fitted with such an arrangement have therefore to be run at slow speeds and lose all the economy resulting from higher speed. Engines whose governors control the valve by positive connection, however, can be run at a much higher speed; there is a certain economical limit to high speed, but up to this point there is nothing to prevent a high piston speed, and secure all the benefits arising therefrom. Nor is it true, as sometimes alleged, that engines of this class—*i. e.*, engines with a posi-

tive movement to the cut-off valve—will not govern closely except at high speeds; because our governor acts equally well at all speeds; it has only to be properly proportioned to its work. A high speed engine would not have just the same governor as one for slow speeds, the latter engine would need more weight in the governor, because centrifugal force varies as the mass moved, and as the square of the velocity; if the velocity is reduced we must increase the weight, and this done there is the same governing power as before.

The governor for a drop cut-off engine has control of the valve only up to the time of releasing it, and during the rest of the stroke the governor does nothing but adjust the tappets properly for the return stroke, any change in speed occurring after cut-off has taken place cannot be provided for until the next stroke, when the point of cut-off is changed according to the speed. With the other form of governor, also, no regulation can take place after cut-off has once occurred until the next stroke, because with any governor the cut-off acts only at the beginning of each stroke; but the positive motion valve, having to open and close the port, and doing it gradually has, for a given cut-off, control during a longer time than the release valve which opens and then instantly closes, so that any change of load would have that much more time to cause a corresponding change in the cut-off. After the cut-off has been accomplished neither governor has any advantage over the other, except that the one directly connected has always control of the valve, and it is sure to be in the proper position each time, which is not certain with the other form.

We have shown that a governor can only change the point of cut-off at the beginning of each stroke, for though it may be sensitive to any variation of load which occurs between the time the valve closes and the end of the stroke, yet it cannot sooner produce any effect. The greater the number of revolutions per minute, the sooner can any change of load be met; this is true of any governor, and it follows that the one which permits the highest speed will have the closest governing, and we have shown that the kind of governor used for a positive valve motion admits of a high speed, and the other variety does not. But it is not necessary to run our governor at a high speed to secure effective close governing, only the point last stated shows that it is capable of the highest results which it is possible to attain. In the article GOVERNOR it is pointed out how extremely sensitive the Cummer governor is at moderate speeds, such as would ordinarily be used, and that it may be relied upon to control the engine under any variation of load pressure or speed likely to occur, or that may be desired. The construction of our governor is such that we can always give such weight to our governor weights as we decide they may need, from the fact that we are not confined in this respect as one would be if the opposing or centripetal force was obtained with a spring.

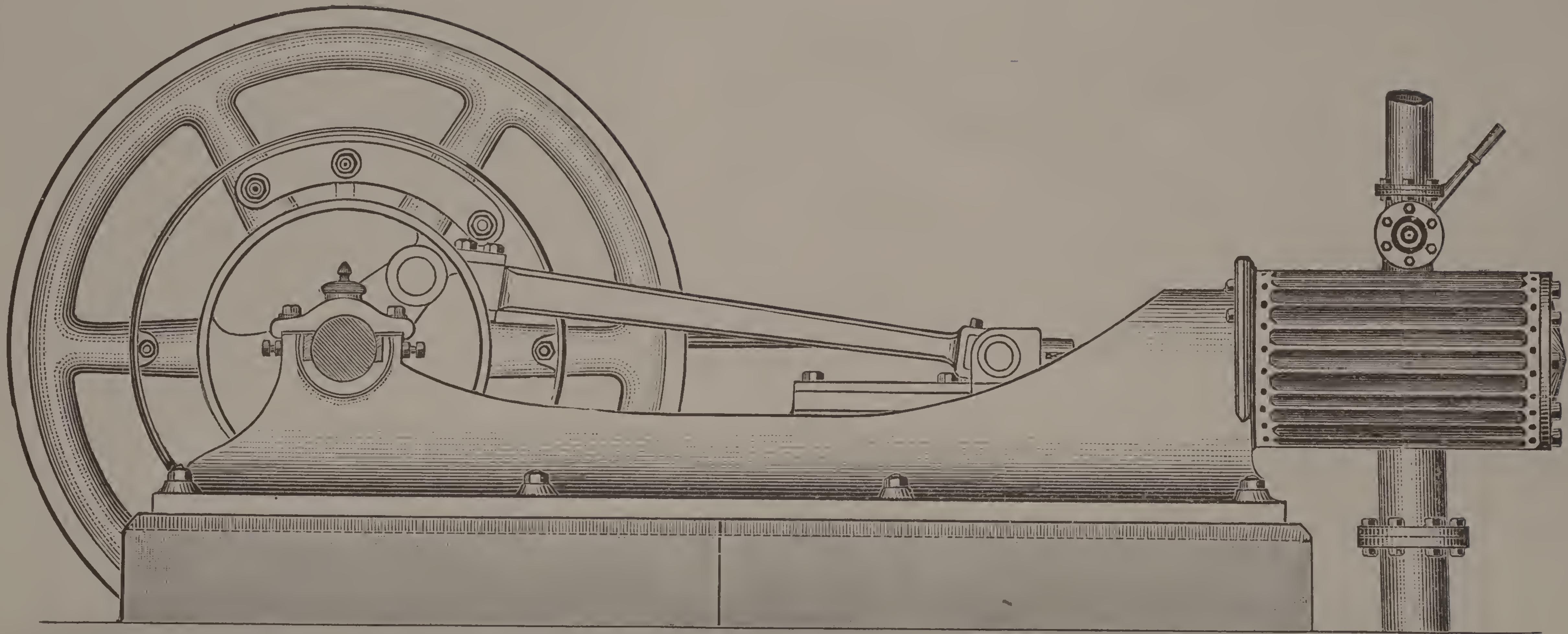


Fig. 2. Elevation of Engine. Class A.

CLASS A. SELF-CONTAINED AUTOMATIC ENGINES.

CYLINDER.		Revolutions per	Piston Speed in	Horse Power Constant for	HORSE POWER NON-CONDENSING WITH 80 lbs. STEAM, CUTTING OFF.				HORSE POWER CONDENSING WITH 80 lbs. STEAM, CUTTING OFF.			
Diameter Inches.	Stroke Inches.	Minute.	Ft. per Min.	1 lb. M. E. P.	At 1-5 Stroke.	At ¼ Stroke. Most economical.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
6	12	200	400	.3427	12 H.P.	14 H. P.	19 H.P.	22 H. P.	16 H. P.	18 H. P.	23 H. P.	26 H. P.
7	12	200	400	.4664	16 "	19 "	25 "	30 "	21 "	25 "	31 "	36 "
8	16	180	480	.7311	24 "	30 "	40 "	47 "	33 "	38 "	49 "	56 "
9	16	180	480	.9252	31 "	37 "	50 "	59 "	42 "	49 "	62 "	70 "

For best economy, all things considered, we recommend 90 lbs. boiler pressure and a cut-off at ¼ stroke. But for cases where a less pressure than 90 lbs. is desired, we have calculated the above table for 80 lbs. pressure. To illustrate the use of the table, suppose we wish to select a non-condensing engine of 20 H. P. Under non-condensing engines in the ¼ stroke column the nearest number is 19 H. P. which is given by a 7x12 engine.

CLASS A. SELF-CONTAINED AUTOMATIC ENGINES.

CYLINDER.		Revolutions per	Piston Speed in	Horse Power Constant for	HORSE POWER NON-CONDENSING WITH 90 lbs. STEAM, CUTTING OFF.				HORSE POWER CONDENSING WITH 90 lbs. STEAM, CUTTING OFF.			
Diameter Inches.	Stroke Inches.	Minute.	Ft. per Min.	1 lb. M. E. P.	At 1-5 Stroke.	At ¼ Stroke. Most economical.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
6	12	200	400	.3427	13 H.P.	16 H. P.	21 H.P.	25 H. P.	17 H. P.	20 H. P.	25 H. P.	29 H. P.
7	12	200	400	.4664	18 "	22 "	29 "	34 "	24 "	27 "	35 "	40 "
8	16	180	480	.7311	28 "	34 "	45 "	53 "	37 "	43 "	54 "	62 "
9	16	180	480	.9252	36 "	43 "	57 "	67 "	47 "	54 "	68 "	78 "

For best economy, all things considered, we recommend 90 lbs. boiler pressure and a cut-off at ¼ stroke. To illustrate the use of the table, suppose we wish to select a non-condensing engine of 20 H. P. Under non-condensing engines in the ¼ stroke column the nearest number is 22 H. P. which is given by a 7x12 engine.

CLASS A. SELF-CONTAINED AUTOMATIC ENGINES.

CYLINDER.		Revolutions per Minute.	Piston Speed in Ft. per Min.	Horse Power Constant for 1 lb. M. E. P.	HORSE POWER NON-CONDENSING WITH 100 lbs. STEAM, CUTTING OFF.				HORSE POWER CONDENSING WITH 100 lbs. STEAM, CUTTING OFF.			
Diameter Inches.	Stroke Inches.				At 1-5 Stroke.	At ¼ Stroke. Most economical.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
6	12	200	400	.3427	15 H.P.	18 H. P.	24 H.P.	28 H. P.	19 H. P.	22 H. P.	28 H. P.	32 H. P.
7	12	200	400	.4664	21 "	24 "	32 "	38 "	26 "	30 "	38 "	43 "
8	16	180	480	.7311	32 "	38 "	51 "	59 "	41 "	47 "	60 "	68 "
9	16	180	480	.9252	41 "	48 "	64 "	75 "	52 "	60 "	75 "	86 "

For best economy, all things considered, we recommend 90 lbs. boiler pressure and a cut-off at ¼ stroke. But for cases where a higher pressure than 90 lbs. is desired, we have calculated the above table for 100 lbs. pressure. To illustrate the use of the table, suppose we wish to select a non-condensing engine of 45 H. P. Under non-condensing engines in the ¼ stroke column the nearest number is 48 H. P. which is given by a 9x16 engine.

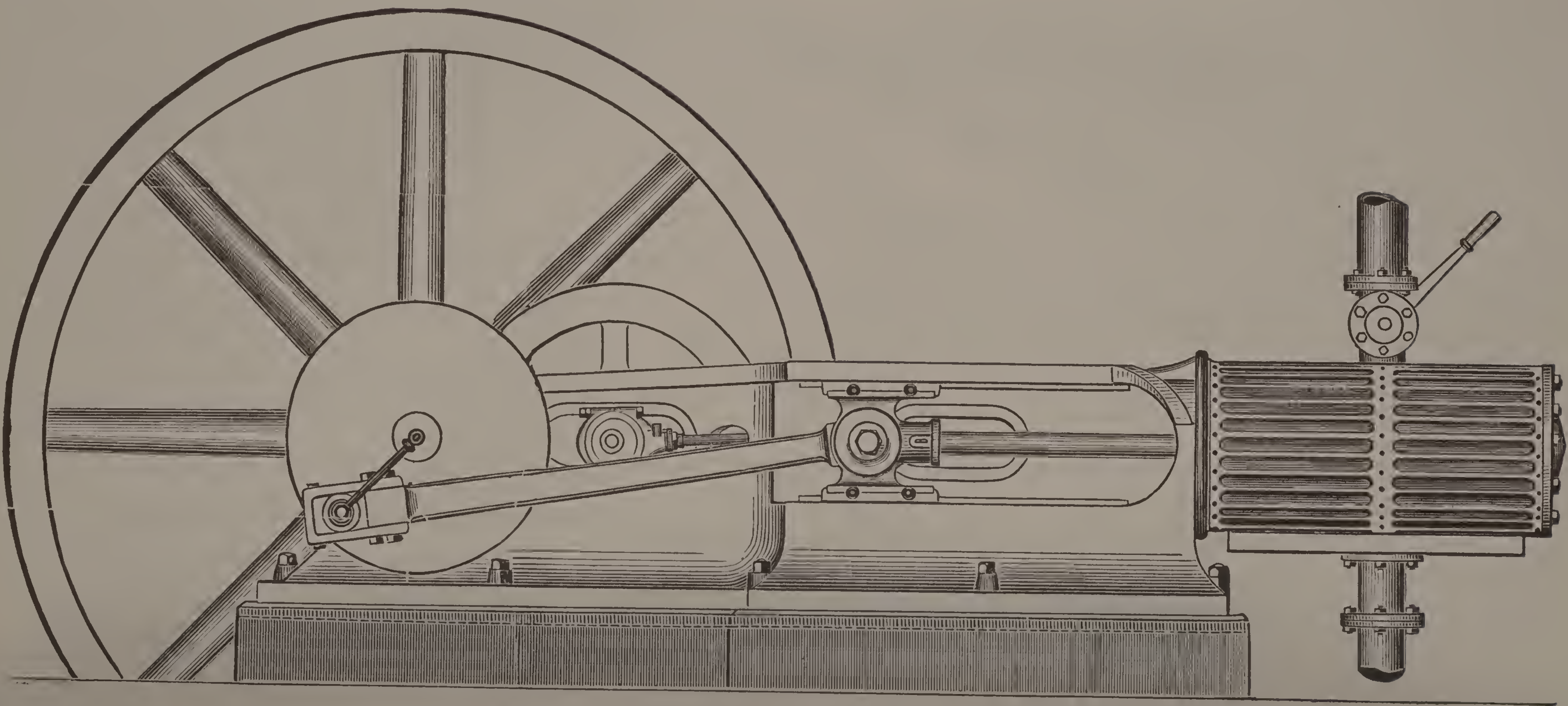


Fig. 3. Elevation of Engine. Class B.

SPECIAL AUTOMATIC ENGINES CLASS B.

Our special engines Class B are designed for moderately large powers, medium high speed, and, like our Class C standard engines, for the highest economy. Since these engines are run at a higher rate of revolution than those of Class C, we have adopted for them a different style of frame in which the guides are supported throughout their whole length, by extending that part of the frame in which they are included, so as to rest upon the foundation, while the cylinder is allowed to overhang. We have also employed a disc crank, which permits better balancing for high speeds than the other form, but otherwise these engines are in every respect precisely like those of Class C, we use the same governor, valves, piston, cross-head and connecting-rod, etc. as with those engines, and the materials and workmanship are equally good. This style of engine is well adapted for its duty and is becoming very popular in many sections of our country.

With respect to the overhanging cylinder, there is no advantage lost by attaching it to the frame in this way, because it has no strains to meet in a vertical direction except its own weight and this is insignificant when the mode of attachment is considered. The end of the girder forms the forward cylinder head, it is turned to fit the counterbore and extends within the cylinder so as to have a bearing of from four to six inches. In this way the bolts connecting the cylinder and frame are relieved from shearing stress, and the weight of the cylinder exerts upon them only a slight tension; while those strains which occur from the steam pressure and from working the engine are all in a horizontal direction, or in line with the axis of the cylinder, and can be fully resisted independently of any support in a vertical direction. With the cross-head and guides the conditions are different, here the strains occur sideways, as well as horizontally, and at high speeds it is especially necessary to make the frame very rigid at the guides to resist the side thrust of the connecting-rod, but at slower speeds this would not be required. Thus our Class C engines have a frame which is unsupported beneath the guides and yet is so rigid at the intended moderate speeds that no vibration whatever occurs. With higher rates of revolution, such as are used with Class B, we must have the guides very rigid, and instead of employing any such make-shift as a central support with a girder frame, we design an entirely new form specially suited to the new conditions which is calculated to resist to the best advantage all such strains as may come upon it and to be very strong and stiff. We embody in this frame the same excellent general features as with Class C, we use the same form of removeable guides, the same main bearing and the same outboard bearing as is employed with those engines; thus for the whole engine we secure special advantages adapting it to the higher speeds

without sacrificing any of the good features of our standard automatic engines.

While this engine is especially adapted for higher speeds than Class C and is so well designed and constructed, that it can closely compete in speed with what are known as "high speed engines;" yet we do not recommend for them speeds much exceeding those given in the tables. A somewhat extended experience in this field, has led us to be opposed to extremely high speed and high speed engines, and we give this opinion for the benefit of our customers; we believe that the popular predudice against very high speed is well founded and that in the future, more moderate rates will be adopted.

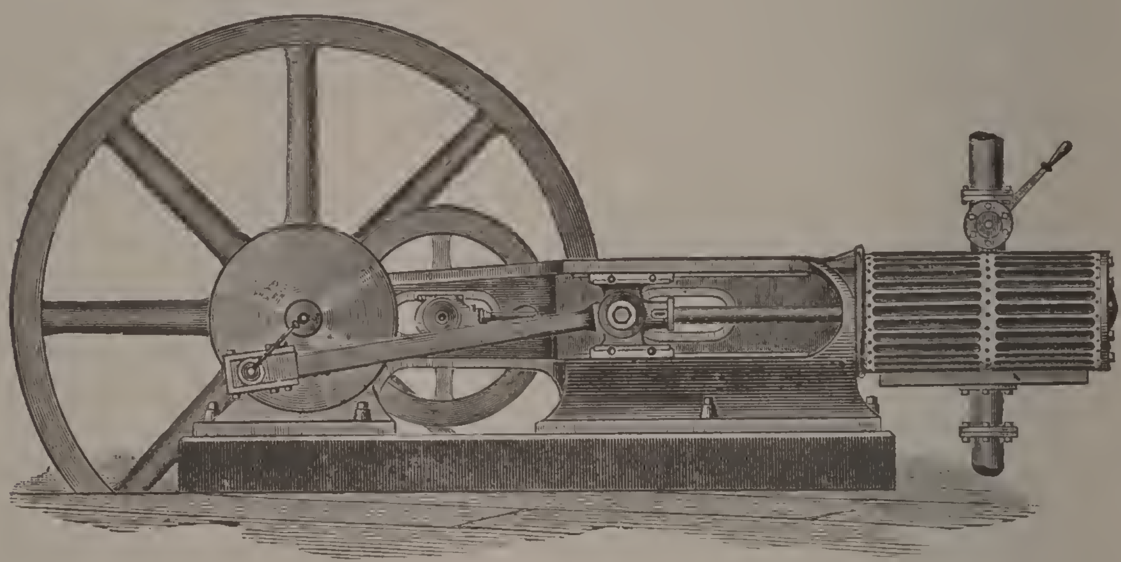


Fig. 3½.

Fig. 3 shows the general features of this engine, but the above small cut is a more correct representation of our later design. The only difference between the two figures, it will be seen, is owing to a change in the form of bed-plate, which, while preserving the same strength and stiffness as before, being well supported beneath the guides and fulfilling the required conditions, is so modified that it allows us to use the same governor for both Class B and Class C. We are thus enabled to make our governors in quantity and have them properly tested before being sent out, and it ensures having them always in stock.

NOTE.—The 10 x 20, 11 x 20 and 12 x 24 engines may, if required, be run at higher rates of speed than the tabular numbers; and, in such cases, we make for them a governor specially adapted for high speeds, which will be furnished with these engines, whenever we are informed that higher speeds are desired.

CLASS B. SPECIAL ENGINES—AUTOMATIC—OVERHANGING CYLINDER.

CYLINDER.		Revolutions per Minute.	Diameter Main Shaft, Inches.	BAND WHEEL.			BELT—(Double.)		FLY-WHEEL.		Diameter Steam Pipe, Inches.	Diameter Exhaust Pipe, Inches.
Diameter, Inches.	Stroke, Inches.			Diameter, Feet.	Face, Inches.	Weight, Pounds.	Width, Inches.	Velocity Ft. per Min.	Diameter, Feet.	Weight, Pounds.		
10	20	150	6	7	11	2800	10	3299	8	3100	3	4
11	20	150	6	7	13	3350	12	3299	8	3700	$3\frac{1}{2}$	5
12	24	140	6	9	13	3400	12	3958	10	3800	4	5
13	24	140	$6\frac{1}{2}$	9	15	3900	14	3958	10	4300	5	6
14	24	140	7	9	18	4550	16	3958	10	5000	5	6
15	30	120	$7\frac{1}{2}$	10	22	7100	20	3769	12	7800	6	7
16	30	120	8	10	24	8200	22	3769	12	9000	6	7
18	34	115	9	12	28	8800	26	4334	13	9700	6	7
20	34	115	10	12	34	10800	32	4334	14	11900	7	8
22	40	100	11	15	38	13200	36	4712	16	14500	7	8
24	40	100	12	15	44	15700	42	4712	16	17300	8	9

The width of belts, as given in the above table, are for Non-Condensing engines. The weights of both fly-wheels and band-wheels are to be considered as approximate only. We will make the wheels as near the given weights as possible.

CLASS B. SPECIAL ENGINES—AUTOMATIC—OVERHANGING CYLINDER.

CYLINDER.		Revolutions	Piston Speed	Horse Power	HORSE POWER NON-CONDENSING WITH 80 lbs. STEAM, CUTTING OFF.				HORSE POWER CONDENSING WITH 80 lbs. STEAM, CUTTING OFF.			
Diameter Inches.	Stroke Inches.	per Minute.	in Ft. per Min.	Constant for 1 lb. M. E. P.	At 1-5 Stroke.	At ¼ Stroke. Most economical.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
10	20	150	500	1.1900	40 H. P.	48 H. P.	65 H. P.	76 H. P.	54 H. P.	62 H. P.	79 H. P.	91 H. P.
11	20	150	500	1.4398	48 "	58 "	78 "	92 "	65 "	76 "	96 "	110 "
12	24	140	560	1.9191	64 "	78 "	105 "	123 "	87 "	101 "	128 "	146 "
13	24	140	560	2.2524	75 "	91 "	123 "	144 "	102 "	118 "	150 "	172 "
14	24	140	560	2.6121	87 "	106 "	142 "	168 "	119 "	137 "	174 "	199 "
15	30	120	600	3.1584	106 "	128 "	172 "	203 "	144 "	166 "	210 "	240 "
16	30	120	600	3.6556	122 "	148 "	199 "	234 "	166 "	192 "	243 "	278 "
18	34	115	650	5.0121	168 "	203 "	273 "	321 "	228 "	263 "	333 "	382 "
20	34	115	650	6.1880	207 "	250 "	337 "	397 "	281 "	325 "	411 "	471 "
22	40	100	667	7.6832	257 "	311 "	418 "	493 "	349 "	403 "	511 "	585 "
24	40	100	667	9.1435	306 "	370 "	498 "	586 "	415 "	480 "	608 "	696 "

For best economy, all things considered, we recommend 90 lbs. boiler pressure and a cut-off at ¼ stroke. But for cases where a less pressure than 90 lbs. is desired we have calculated the above table for 80 lbs. pressure. To illustrate the use of the table, suppose we wish to select a non-condensing engine of 75 H. P. Under non-condensing engines, in the ¼ stroke column, the nearest number is 78 H. P., which is given by a 12x24 engine. If a condensing engine is desired, we can get the same power from a smaller engine and smaller boiler than is required with a non-condensing engine. In most cases the reduction in price from this cause will equal, and sometimes more than equal, the cost of a condenser, while the purchaser has afterwards the advantage of increased economy in fuel.

CLASS B. SPECIAL ENGINES—AUTOMATIC—OVERHANGING CYLINDER.

CYLINDER.		Revolutions per	Piston Speed in	Horse Power Constant for	HORSE POWER NON-CONDENSING WITH 90 lbs. STEAM, CUTTING OFF.				HORSE POWER CONDENSING WITH 90 lbs. STEAM, CUTTING OFF.			
Diameter Inches.	Stroke Inches.	Minute.	Ft. per Min.	1 lb. M. E. P.	At 1-5 Stroke.	At ¼ Stroke. Most economical.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
10	20	150	500	1.1900	46 H. P.	55 H. P.	74 H. P.	86 H. P.	60 H. P.	70 H. P.	88 H. P.	101 H. P.
11	20	150	500	1.4398	56 "	67 "	89 "	105 "	73 "	84 "	106 "	122 "
12	24	140	560	1.9191	74 "	89 "	119 "	139 "	97 "	112 "	142 "	162 "
13	24	140	560	2.2524	87 "	105 "	139 "	164 "	114 "	132 "	167 "	191 "
14	24	140	560	2.6121	101 "	121 "	162 "	190 "	132 "	153 "	193 "	221 "
15	30	120	600	3.1584	122 "	147 "	196 "	229 "	160 "	185 "	233 "	267 "
16	30	120	600	3.6556	141 "	170 "	226 "	265 "	185 "	214 "	270 "	309 "
18	34	115	650	5.0121	194 "	233 "	310 "	364 "	254 "	293 "	370 "	424 "
20	34	115	650	6.1880	239 "	287 "	383 "	449 "	313 "	361 "	457 "	523 "
22	40	100	667	7.6832	297 "	357 "	476 "	558 "	389 "	449 "	568 "	650 "
24	40	100	667	9.1435	353 "	424 "	566 "	664 "	463 "	534 "	676 "	773 "

For best economy, all things considered, we recommend 90 lbs. boiler pressure and a cut-off at ¼ stroke. To illustrate the use of the table, sup- we wish to select a non-condensing engine of 100 H. P. Under non-condensing engines, in the ¼ stroke column, the nearest number is 105 H. P., which is given by a 13x24 engine. If a condensing engine is desired, we can get the same power from a smaller engine and smaller boiler than is required with a non-condensing engine. In most cases the reduction in price from this cause will equal, and sometimes more than equal, the cost of a condenser, while the purchaser has afterwards the advantage of increased economy in fuel.

CLASS B. SPECIAL ENGINES—AUTOMATIC—OVERHANGING CYLINDER.

CYLINDER.		Revolutions per	Piston Speed in	Horse Power Constant for	HORSE POWER NON-CONDENSING WITH 100 lbs. STEAM, CUTTING OFF				HORSE POWER CONDENSING WITH 100 lbs. STEAM, CUTTING OFF			
Diameter Inches.	Stroke Inches.	Minute.	Ft. per Min.	1 lb. M. E. P.	At 1-5 Stroke.	At ¼ Stroke. Most economical.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
10	20	150	500	1.1900	52 H. P.	62 H. P.	83 H. P.	96 H. P.	67 H. P.	77 H. P.	97 H. P.	111 H. P.
11	20	150	500	1.4398	63 "	75 "	100 "	117 "	80 "	93 "	117 "	134 "
12	24	140	560	1.9191	84 "	101 "	133 "	156 "	107 "	124 "	156 "	179 "
13	24	140	560	2.2524	99 "	118 "	156 "	183 "	126 "	145 "	183 "	210 "
14	24	140	560	2.6121	115 "	137 "	181 "	212 "	146 "	168 "	213 "	243 "
15	30	120	600	3.1584	139 "	165 "	219 "	256 "	177 "	203 "	257 "	294 "
16	30	120	600	3.6556	160 "	191 "	254 "	296 "	204 "	235 "	297 "	340 "
18	34	115	650	5.0121	220 "	262 "	348 "	406 "	280 "	323 "	408 "	466 "
20	34	115	650	6.1880	272 "	324 "	429 "	502 "	346 "	398 "	503 "	576 "
22	40	100	667	7.6832	337 "	402 "	533 "	623 "	429 "	495 "	625 "	715 "
24	40	100	667	9.1435	401 "	479 "	634 "	741 "	511 "	589 "	744 "	851 "

For best economy, all things considered, we recommend 90 lbs. boiler pressure and a cut-off at ¼ stroke. But for cases where a higher pressure than 90 lbs. is desired, we have calculated the above table for 100 lbs. pressure. To illustrate the use of the table, suppose we wish to select a non-condensing engine of 130 H. P. Under non-condensing engines, in the ¼ stroke column, the nearest number is 137 H. P., which is given by a 14x24 engine. If a condensing engine is desired, we can get the same power from a smaller engine and smaller boiler than is required with a non-condensing engine. In most cases the reduction in price from this cause will equal, and sometimes more than equal, the cost of a condenser, while the purchaser has afterwards the advantage of increased economy in fuel.

INDEPENDENT CONDENSING APPARATUS

As Adapted for CLASS B SPECIAL ENGINES.

CYLINDER.		Revolutions per	Piston Speed in	Horse Power Constant for	HORSE POWER CONDENSING WITH 90 lbs. STEAM, CUTTING-OFF.				SIZE OF CONDENSER REQUIRED FOR DIFFERENT POINTS OF CUT-OFF.			
Diameter, Inches.	Stroke, Inches.	Minute.	Ft. per Min.	1 lb. M. E. P.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
10	20	150	500	1.1900	60 H. P.	70 H. P.	88 H. P.	101 H. P.			A	A
11	20	150	500	1.4398	73 "	84 "	106 "	122 "		A	A	B
12	24	140	560	1.9191	97 "	112 "	142 "	162 "	A	A	B	C
13	24	140	560	2.2524	114 "	132 "	167 "	191 "	A	A	B	C
14	24	140	560	2.6121	132 "	153 "	193 "	221 "	A	B	C	C
15	30	120	600	3.1584	160 "	185 "	233 "	267 "	B	B	C	D
16	30	120	600	3.6556	185 "	214 "	270 "	309 "	B	C	D	D
18	34	115	650	5.0121	254 "	293 "	370 "	424 "	C	D	E	F
20	34	115	650	6.1880	313 "	361 "	457 "	523 "	C	D	E	G
22	40	100	667	7.6832	389 "	449 "	568 "	650 "	D	E	F	G
24	40	100	667	9.1435	463 "	534 "	676 "	773 "	D	E	F	H

The power ratings in the above table are based on a boiler pressure of 90 lbs., but the same condensers as are given in this table for the various engines and points of cut-off may be used also when 80 or 100 lbs. steam pressure is employed. The sizes of condensers are denoted by letters as A, B, C &c., and by reference to the table of Independent Condensing Apparatus, all the necessary dimensions, corresponding to any given size, may be readily ascertained.

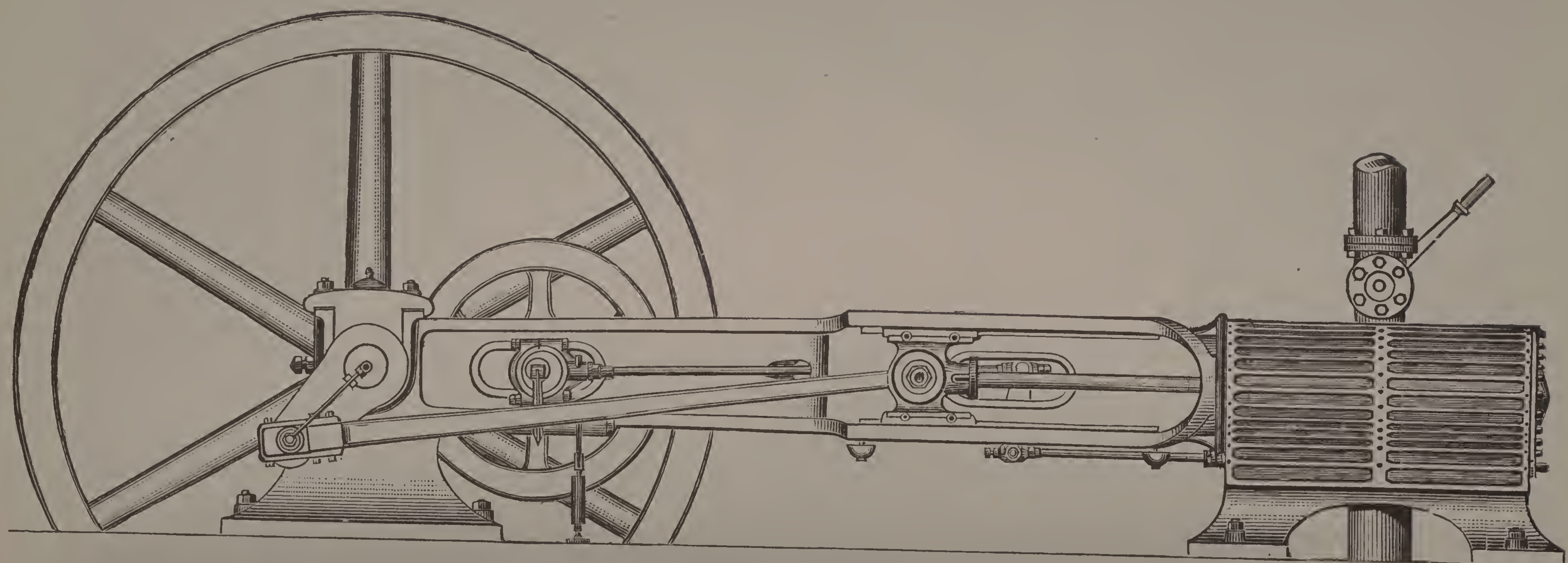


Fig. 4. Elevation of Engine. Class C.

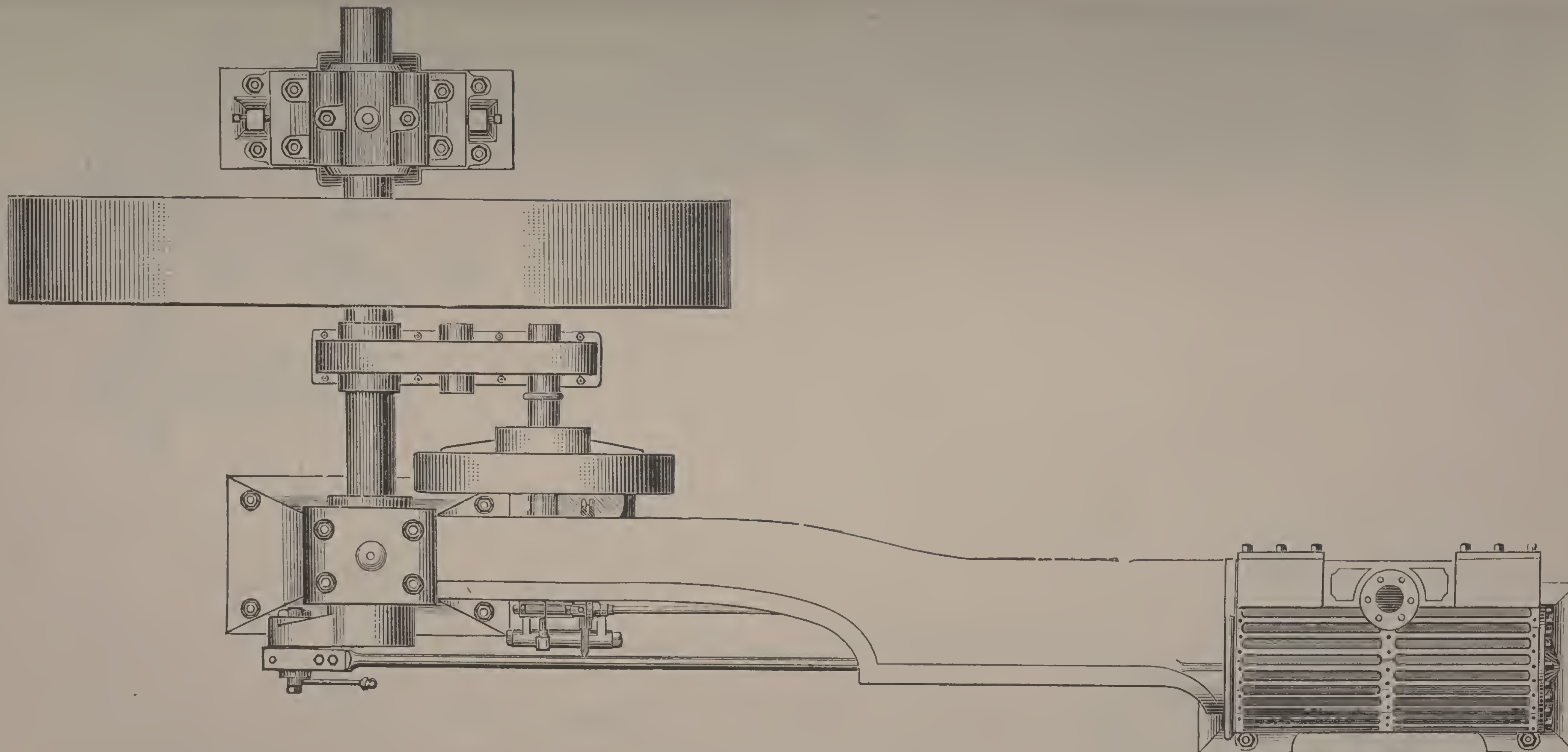


Fig 5. Plan of Engine. Class C.

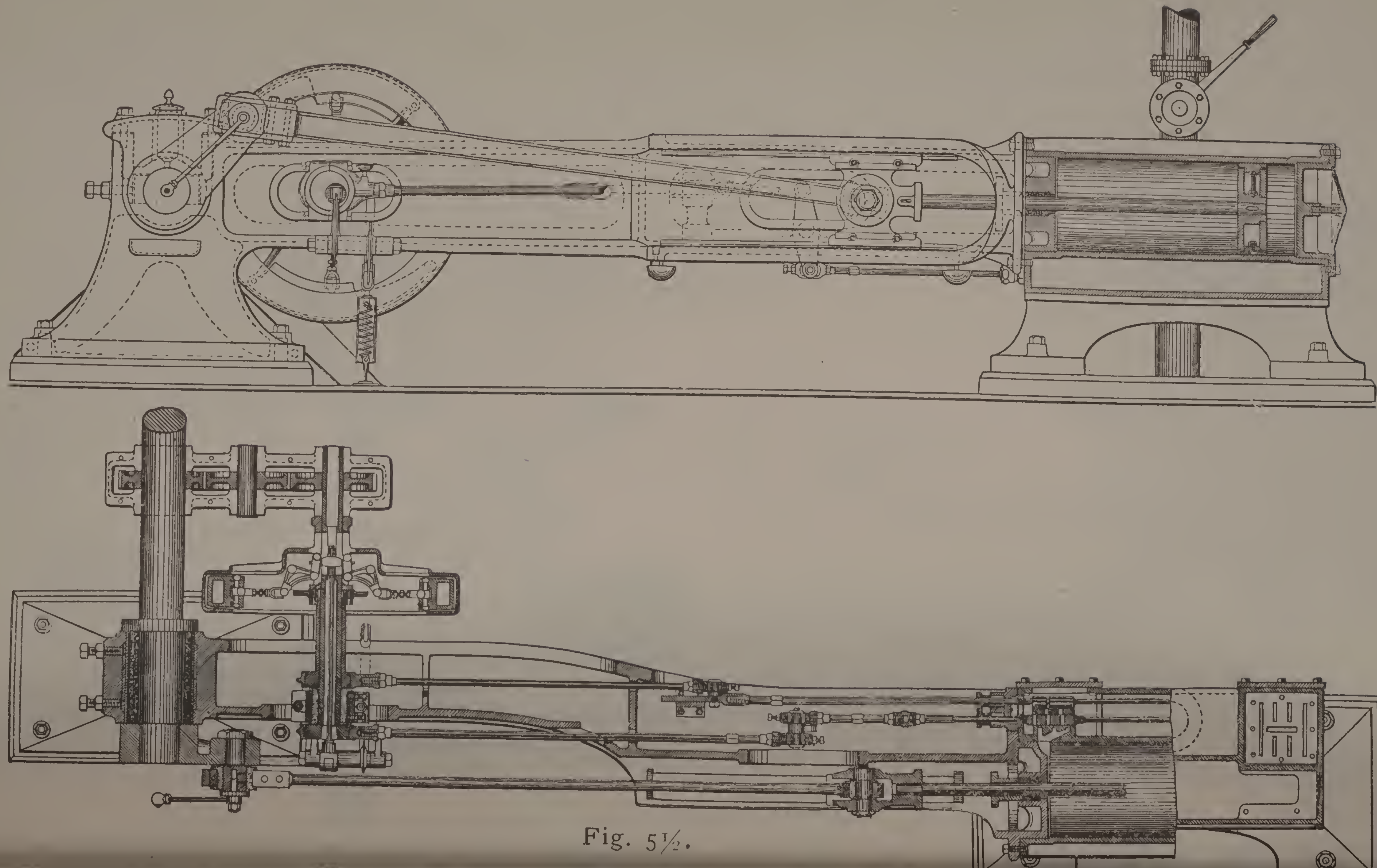


Fig. 5½.

GENERAL DESCRIPTION OF CLASS C ENGINE.

Fig. 5½ shows an elevation and part vertical section of a Class C standard engine, and also a horizontal section which shows the form of the girder and cylinder, and clearly exhibits the various working parts, in their relations to each other. In the elevation there appears a section of the piston and there is shown our form of cross-head, connecting-rod and crank. The horizontal section further details the cross-head, the stub ends of the connecting-rod and the crank. The main bearing with its quarter boxes, shoes and means for adjustment is clearly detailed and there is shown the train of three gears which drives the governor shaft from the main shaft; the governor also appears in section showing the weights, bell cranks and attachments to the thrust-rod, which are all similarly illustrated in Fig. 6 and explained in the description of the governor. The elevation and plan exhibits the large bell crank, to the vertical arm of which the thrust-rod is attached, while the horizontal arm supports the hanging weights and has attached to it the spiral spring which appears in the elevation; all these parts are also clearly seen in Fig. 6. Referring once more to the horizontal section it will be noticed that the main eccentric is attached directly to the governor shaft, and the cut-off eccentric is attached to a long sleeve, which by suitable connection with the governor weights, is made to rotate the eccentric forward or backward and thus change the point of cut-off. The main eccentric operates the main valves and also the exhaust valves through the intervention of a rock arm. The eccentric rods and valve-stems have means for adjustment of length, these with the joints at the rock arm and the slide to prevent vibration of the cut-off valve-stem, as well as the mode for attaching the valves to the valve-stems are all shown in the horizontal section.

The lower right hand portion of this section, shows that part of the cylinder where the exhaust valve for this end is situated. The piston-rod stuffing box is bushed with brass; we use for the piston-rod and also for the valve-stems an improved form of metallic packing which will be found illustrated and described in another place. In the vertical section appears our form of cylinder head which projects within the cylinder and reduces clearance to as low an amount as possible. The forward head is formed by the end of the girder frame, but in our larger sizes the head is made separate and bolted to the frame. Objections have been made against this deep form of cylinder head, on the ground that the annular space between the head and counterbore affords so much more condensing surface than the ordinary form of cylinder head; but, in point of fact, this objection does not hold at all, because we make our cylinder heads a true fit for an inch or so at the outside bearing and an easy sliding fit for the rest of the length. The space between the counterbore and head, which is about the thickness of a piece of paper, becomes when in use, completely filled up with a deposit from the oil used to lubricate the cylinder which is driven in by the steam and effectually prevents any condensation from this cause. This we know to be the fact from our own experience and if it were not, we could easily devise means to overcome what would be, if it really existed, a valid objection against this form of cylinder head. The various details alluded to in this general description, will be found more thoroughly illustrated and described in other portions of this catalogue.

CLASS C. STANDARD ENGINES—AUTOMATIC. GIRDER FRAME.

CYLINDER.		Revolutions per	Diameter Main	BAND WHEEL.			BELT—(Double.)		FLY-WHEEL.		Diameter Steam	Diameter Exhaust
Diameter, Inches.	Stroke, Inches.	Minute.	Shaft, Inches.	Diameter, Feet.	Face, Inches.	Weight, Pounds.	Width, Inches.	Velocity Ft. per Min.	Diameter, Feet.	Weight, Pounds.	Pipe, Inches.	Pipe, Inches.
12	30	120	6	10	16	5168	14	3769	12	5800	4	5
14	30	120	7	10	20	6977	18	3769	12	7700	5	6
16	36	100	8	14	22	7900	20	4398	16	8700	6	7
18	36	100	9	14	26	9925	24	4398	16	10900	6	7
20	42	93	10	16	32	12700	30	4674	18	14000	7	8
22	42	93	11	16	38	15200	36	4674	18	16700	7	8
24	48	81	12	18	44	21550	42	4580	20	23700	8	9
26	48	81	12½	18	50	25000	48	4580	20	27500	8	10
							TWO.					
28	48	81	13	18	60	28800	28	4580	20	31700	9	10
30	48	81	13½	18	68	32800	32	4580	20	36100	10	11
32	48	81	14	18	80	37000	38	4580	20	40700	10	11
34	48	81	14½	18	88	41350	42	4580	20	45500	11	12
36	48	81	15	18	102	45800	48	4580	20	50400	11	12
							THREE.					
38	48	81	15½	18	114	50700	36	4580	20	55800	12	14
40	48	81	16	18	126	55850	40	4580	20	61400	12	14

The width of belts, as given in the above table, are for Non-Condensing engines. The weights of both fly-wheels and band-wheels are to be considered as approximate only. We will make the wheels as near the given weights as possible.

CLASS C. STANDARD ENGINES—AUTOMATIC. GIRDER FRAME.

CYLINDER.		Revolutions per Minute.	Piston Speed in Ft. per Min.	Horse Power Constant for 1 lb. M. E. P.	HORSE POWER NON-CONDENSING WITH 80 lbs. STEAM, CUTTING OFF.				HORSE POWER CONDENSING WITH 80 lbs. STEAM, CUTTING OFF.			
Diameter Inches.	Stroke Inches.				At 1-5 Stroke.	At ¼ Stroke. Most economical.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
12	30	120	600	2.0562	69 H. P.	83 H. P.	112 H. P.	132 H. P.	93 H. P.	108 H. P.	137 H. P.	157 H. P.
14	30	120	600	2.7987	94 "	113 "	152 "	180 "	127 "	147 "	186 "	213 "
16	36	100	600	3.6556	122 "	148 "	199 "	234 "	166 "	192 "	243 "	278 "
18	36	100	600	4.6265	155 "	187 "	252 "	297 "	210 "	243 "	308 "	352 "
20	42	93	650	6.1880	207 "	250 "	337 "	397 "	281 "	325 "	411 "	471 "
22	42	93	650	7.4875	250 "	303 "	408 "	480 "	340 "	393 "	498 "	570 "
24	48	81	650	8.9105	298 "	360 "	485 "	571 "	405 "	467 "	592 "	678 "
26	48	81	650	10.4398	349 "	422 "	569 "	669 "	474 "	548 "	694 "	795 "
28	48	81	650	12.1284	406 "	491 "	661 "	778 "	551 "	636 "	806 "	923 "
30	48	81	650	13.9228	465 "	563 "	758 "	893 "	633 "	730 "	925 "	1060 "
32	48	81	650	15.8411	530 "	641 "	863 "	1016 "	720 "	831 "	1053 "	1206 "
34	48	81	650	17.8833	598 "	723 "	974 "	1147 "	812 "	938 "	1189 "	1361 "
36	48	81	650	20.0489	670 "	811 "	1092 "	1286 "	911 "	1051 "	1332 "	1526 "
38	48	81	650	22.3385	747 "	903 "	1217 "	1432 "	1015 "	1171 "	1485 "	1700 "
40	48	81	650	24.7520	828 "	1001 "	1348 "	1587 "	1125 "	1298 "	1645 "	1884 "

For best economy, all things considered, we recommend 90 lbs. boiler pressure and a cut-off at ¼ stroke. But for cases where a less pressure than 90 lbs. is desired we have calculated the above table for 80 lbs. pressure. To illustrate the use of the table, suppose we wish to select a non-condensing engine of 180 H. P. Under non-condensing engines, in the ¼ stroke column, the nearest number is 187 H. P. which is given by an 18x36 engine. If a condensing engine is desired, we can get the same power from a smaller engine and smaller boiler than is required with a non-condensing engine. In most cases the reduction in price from this caus.. will equal, and sometimes more than equal, the cost of a condenser, while the purchaser has afterwards the advantage of increased economy in fuel.

CLASS C. STANDARD ENGINES—AUTOMATIC. GIRDER FRAME.

CYLINDER.		Revolutions	Piston Speed	Horse Power	HORSE POWER NON-CONDENSING WITH 90 lbs. STEAM, CUTTING OFF				HORSE POWER CONDENSING WITH 90 lbs. STEAM, CUTTING OFF			
Diameter, Inches.	Stroke, Inches.	per Minute.	in Ft. per Min.	Constant for 1 lb. M. E. P.	At 1-5 Stroke.	At ¼ Stroke. Most economical.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
12	30	120	600	2.0562	80 H. P.	95 H. P.	127 H. P.	149 H. P.	104 H. P.	120 H. P.	152 H. P.	172 H. P.
14	30	120	600	2.7987	108 "	130 "	173 "	203 "	142 "	163 "	207 "	337 "
16	36	100	600	3.6556	141 "	170 "	226 "	265 "	185 "	214 "	270 "	309 "
18	36	100	600	4.6265	179 "	215 "	286 "	336 "	234 "	270 "	342 "	391 "
20	42	93	650	6.1880	239 "	287 "	383 "	449 "	313 "	361 "	457 "	523 "
22	42	93	650	7.4875	289 "	347 "	464 "	543 "	379 "	437 "	553 "	633 "
24	48	81	650	8.9105	344 "	414 "	552 "	647 "	451 "	520 "	659 "	754 "
26	48	81	650	10.4398	404 "	484 "	646 "	758 "	529 "	610 "	772 "	883 "
28	48	81	650	12.1284	469 "	563 "	751 "	880 "	614 "	708 "	896 "	1026 "
30	48	81	650	13.9228	538 "	646 "	862 "	1011 "	705 "	813 "	1029 "	1178 "
32	48	81	650	15.8411	612 "	735 "	981 "	1150 "	802 "	925 "	1171 "	1340 "
34	48	81	650	17.8833	691 "	830 "	1107 "	1298 "	906 "	1044 "	1322 "	1513 "
36	48	81	650	20.0489	775 "	930 "	1241 "	1455 "	1016 "	1171 "	1482 "	1696 "
38	48	81	650	22.3385	863 "	1037 "	1383 "	1621 "	1132 "	1305 "	1651 "	1889 "
40	48	81	650	24.7520	957 "	1149 "	1532 "	1797 "	1254 "	1446 "	1829 "	2094 "

For best economy, all things considered, we recommend 90 lbs. boiler pressure and a cut-off at ¼ stroke. To illustrate the use of the table, suppose we wish to select a non-condensing engine of 200 H. P Under non-condensing engines, in the ¼ stroke column, the nearest number is 215 H. P. which is given by an 18x36 engine. If a condensing engine is desired, we can get the same power from a smaller engine and smaller boiler than is required with a non-condensing engine. In most cases the reduction in price from this cause will equal, and sometimes more than equal, the cost of a condenser, while the purchaser has afterwards the advantage of increased economy in fuel.

CLASS C. STANDARD ENGINES—AUTOMATIC. GIRDER FRAME.

CYLINDER.		Revolutions per	Piston Speed in	Horse Power Constant for	HORSE POWER NON-CONDENSING WITH 100 lbs. STEAM, CUTTING OFF				HORSE POWER CONDENSING WITH 100 lbs. STEAM, CUTTING OFF			
Diameter, Inches.	Stroke, Inches.	Minute.	Ft. per Min.	1 lb. M. E. P.	At 1-5 Stroke.	At ¼ Stroke. Most economical.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
12	30	120	600	2.0562	90 H. P.	108 H. P.	143 H. P.	167 H. P.	115 H. P.	132 H. P.	167 H. P.	191 H. P.
14	30	120	600	2.7987	123 "	147 "	194 "	227 "	156 "	180 "	228 "	260 "
16	36	100	600	3.6556	160 "	191 "	254 "	296 "	204 "	235 "	297 "	340 "
18	36	100	600	4.6265	203 "	242 "	321 "	375 "	259 "	298 "	376 "	430 "
20	42	93	650	6.1880	272 "	325 "	429 "	502 "	346 "	398 "	503 "	576 "
22	42	93	650	7.4875	329 "	392 "	519 "	607 "	418 "	482 "	609 "	697 "
24	48	81	650	8.9105	391 "	467 "	618 "	722 "	498 "	574 "	725 "	829 "
26	48	81	650	10.4398	458 "	547 "	724 "	846 "	583 "	672 "	849 "	971 "
28	48	81	650	12.1284	532 "	635 "	841 "	983 "	678 "	781 "	987 "	1128 "
30	48	81	650	13.9228	611 "	729 "	965 "	1128 "	778 "	896 "	1133 "	1295 "
32	48	81	650	15.8411	695 "	829 "	1098 "	1284 "	885 "	1020 "	1279 "	1474 "
34	48	81	650	17.8833	785 "	936 "	1240 "	1449 "	999 "	1151 "	1455 "	1664 "
36	48	81	650	20.0489	890 "	1050 "	1390 "	1625 "	1120 "	1290 "	1631 "	1865 "
38	48	81	650	22.3385	980 "	1170 "	1549 "	1810 "	1248 "	1438 "	1817 "	2078 "
40	48	81	650	24.7520	1086 "	1296 "	1716 "	2006 "	1383 "	1593 "	2013 "	2303 "

For best economy, all things considered, we recommend 90 lbs. boiler pressure and a cut-off at ¼ stroke. But for cases where a higher pressure than 90 lbs. is desired we have calculated the above table for 100 lbs. pressure. To illustrate the use of the table, suppose we wish to select a non-condensing engine of 240 H. P. Under non-condensing engines, in the ¼ stroke column, the nearest number is 242 H. P. which is given by an 18x36 engine. If a condensing engine is desired, we can get the same power from a smaller engine and smaller boiler than is required with a non-condensing engine. In most cases the reduction in price from this cause will equal, and sometimes more than equal, the cost of a condenser, while the purchaser has afterwards the advantage of increased economy in fuel.

INDEPENDENT CONDENSING APPARATUS

As Adapted for CLASS C STANDARD ENGINES.

CYLINDER.		Revolutions per	Piston Speed in	Horse Power Constant for	HORSE POWER CONDENSING WITH 90 lbs. STEAM, CUTTING-OFF				SIZE OF CONDENSER REQUIRED FOR DIFFERENT POINTS OF CUT-OFF.			
Diameter, Inches.	Stroke, Inches.	Minute.	Ft. per Min.	1 lb. M. E. P.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.	At 1-5 Stroke.	At ¼ Stroke.	At ⅜ Stroke.	At ½ Stroke.
12	30	120	600	2.0562	104H.P.	120H.P.	152H.P.	172H.P.	A	A	B	C
14	30	120	600	2.7987	142 "	163 "	207 "	337 "	A	B	C	D
16	36	100	600	3.6556	185 "	219 "	270 "	309 "	B	B	D	E
18	36	100	600	4.6265	234 "	270 "	342 "	391 "	B	C	D	E
20	42	93	650	6.1880	313 "	361 "	457 "	523 "	C	D	E	G
22	42	93	650	7.4885	379 "	437 "	553 "	633 "	D	E	F	H
24	48	81	650	8.9105	452 "	520 "	659 "	754 "	D	E	G	H
26	48	81	650	10.4398	529 "	610 "	772 "	883 "	E	F	H	I
28	48	81	650	12.1284	614 "	708 "	896 "	1026 "	F	G	H	J
30	48	81	650	13.9228	705 "	813 "	1029 "	1178 "	F	G	I	K
32	48	81	650	15.8411	802 "	925 "	1171 "	1340 "	G	H	I	L
34	48	81	650	17.8813	906 "	1044 "	1322 "	1513 "	G	H	I	L
36	48	81	650	20.0489	1016 "	1171 "	1482 "	1696 "	H	I	K	L
38	48	81	650	22.3385	1132 "	1305 "	1651 "	1889 "	H	I	K	L
40	48	81	650	24.7520	1254 "	1446 "	1829 "	2094 "	I	J	L	M

The power ratings in the above table are based on a boiler pressure of 90 lbs., but the same condensers as are given in this table for the various engines and points of cut-off may be used also when 80 or 100 lbs. steam pressure is employed. The sizes of condensers are denoted by letters as A, B, C &c., and by reference to the table of Independent Condensing Apparatus, all the necessary dimensions, corresponding to any given size, may be readily ascertained.

DESCRIPTION OF GOVERNOR.

Fig. 6 is a vertical section through the center of the governor and its parts, together with a cross section of the girder of the engine; this section being along the center line of the governor shaft shows the main eccentric cast solid with it. The cut-off eccentric, with its sleeve, it will be observed, fits loosely on the governor shaft, and is connected with the

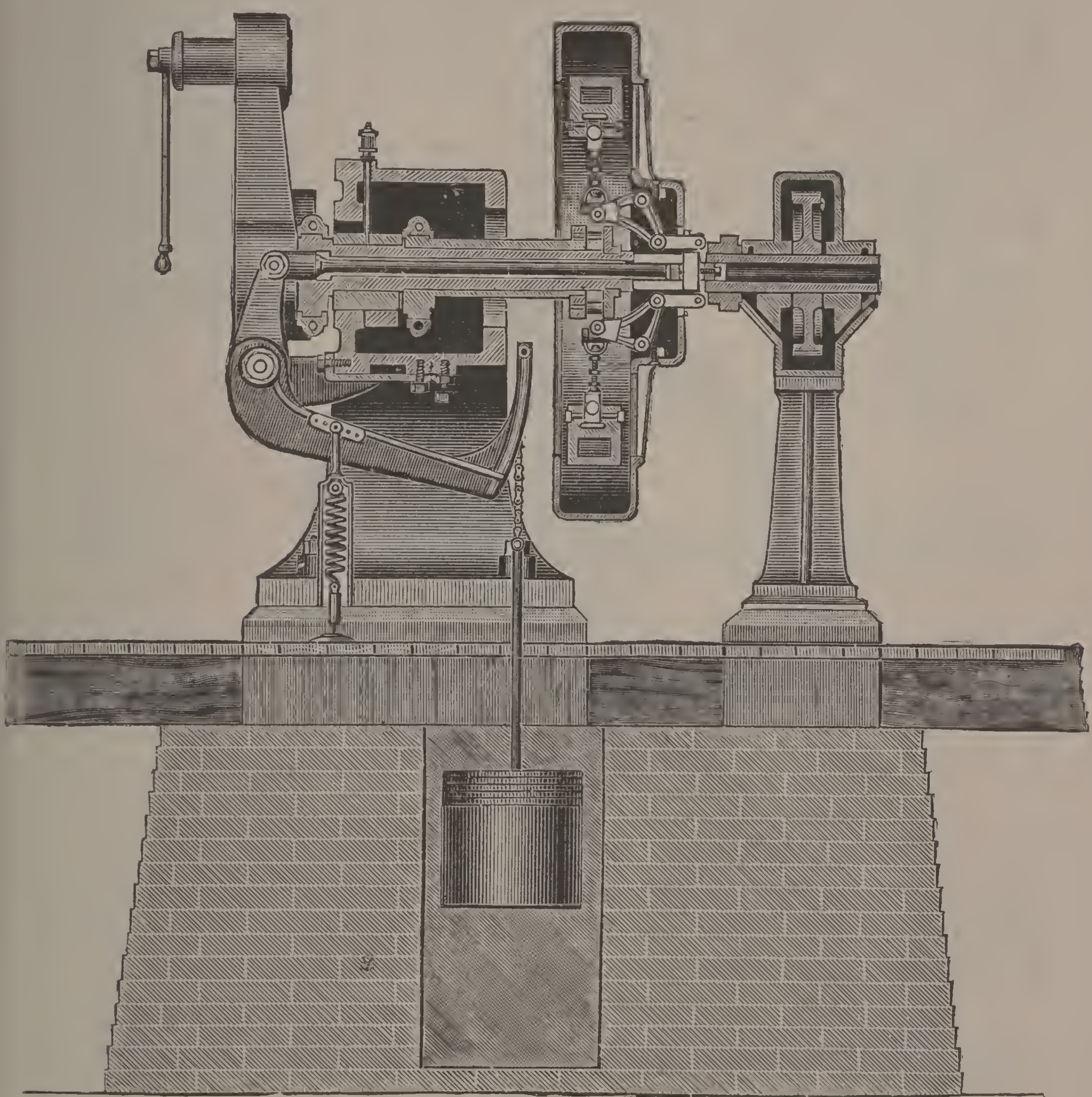


Fig. 6.

flying ends of the governor weights by means of rods or links (as shown in Fig. 7) in such a manner that the cut-off eccentric, with its sleeve, is moved around the governor shaft, either forward or backward, as the flying weights change their position; by this means, the steam is cut off correspondingly earlier or later in the stroke as the governor or flying weights adjust themselves to the load.

The governor shaft is driven from the main shaft by a train of gears, one of which appears in section in Fig. 6. The governor case, to which is attached the flying weights, is keyed to the governor shaft, and revolves with it. It will be noticed the governor shaft is hollow, and has passing through it a thrust rod. One end of this thrust rod is attached to a cross bar, which, passing through a slot in the go rnor shaft, is thereby made to revolve with it. The cross bar just referred to is connected with the governor or flying weights by suitable connections and bell cranks shown in Fig. 6, and further detailed in Fig. 8. The other end of the thrust rod fits into a step which is jointed to the vertical arm of the large bell crank shown in Fig. 6; it will be clear that any movement of the weights in the governor case would cause the thrust rod to move correspondingly out or in, and thus operate or change the relative position of the large bell crank and cause the weight located under the

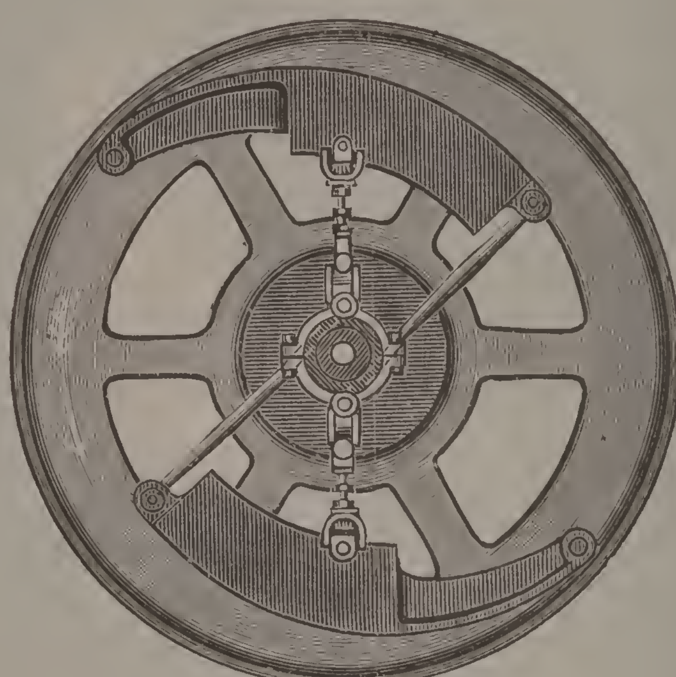


Fig. 7.

engine and attached to the end of the horizontal arm of the bell crank (as shown in Fig. 6) to be raised or lowered in an amount corresponding to the outer or inner position of the governor weights.

Referring again to the cut-off eccentric in Fig. 6, which shows it attached to a long sleeve, also seen in Fig. 7; in the latter engraving it is shown how the flying ends of the governor weights are connected by means of two rods and a clamp collar, with the sleeve of the cut-off eccentric so that, as the governor weights change their position, the eccentric, with its sleeve, moves around the shaft either forward or backward. When the cut-off eccentric is rotated forward the steam is cut off earlier in the stroke; when the eccentric is rotated backward, the steam is cut off later in the stroke. The extreme range of cut-off as controlled by this governor may be from o to 8–10 of the stroke

measured from the beginning ; these extremes correspond to the extreme positions of the flying weights, or, in other words, the engine is controlled by the governor from a simple friction load to the full capacity of the engine. It will be seen that the dead weights suspended from the horizontal arm of the large bell crank, can be varied or adjusted in amount. This provision is made in order to regulate the speed of the engine. Whenever a change, either faster or slower, than standard speed is desired, the required variation is effected by simply adding to or taking from these loose weights under the bed ; the change is easily made without the necessity for stopping the engine.

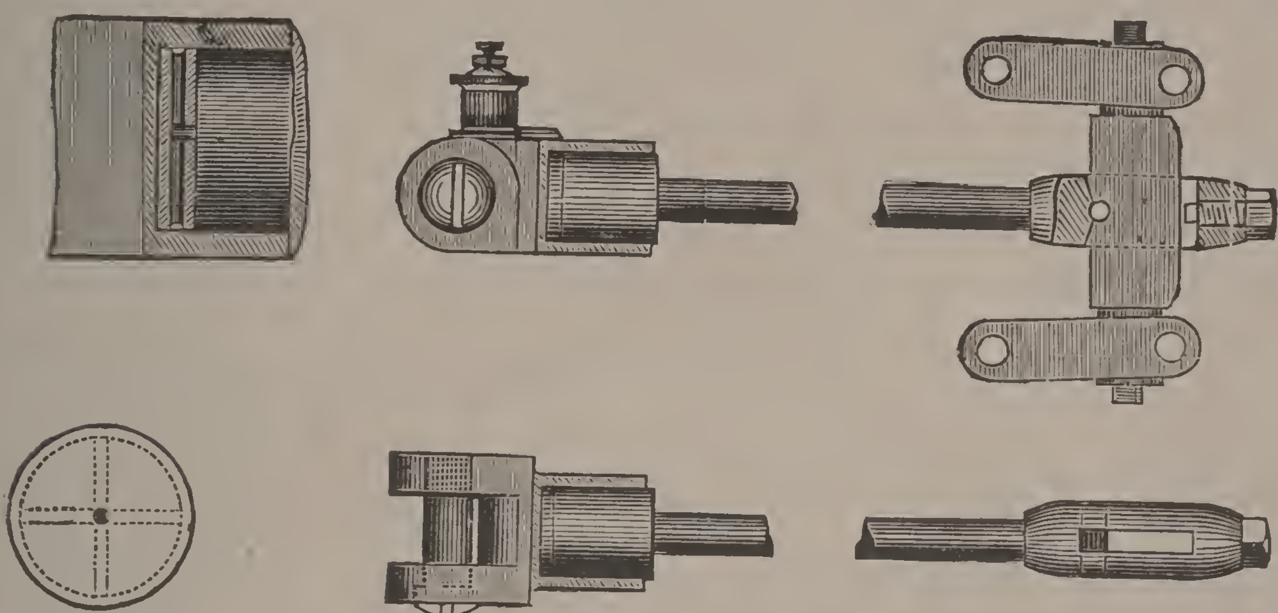


Fig. 8.

The above figure represents the thrust rod and connections and pivotal step, the rod is shown broken to save space. The rod in the upper figure is sectioned at one end to show the cross bar and links and at the other, to show that part of the pivotal joint which forms a case for the step and the end of the thrust rod. Immediately below is a plan of these same parts, and at the extreme left is an enlarged section of the step and part of its case ; the step also appears in plan ; the holes shown in the section and by dotted lines in the plan exhibits the mode of oiling the step by means of an oil cup which also oils thc pivot.

The whole mechanism, described above at length, makes a positive connection for the governor ; it has no belt to slip or be thrown off, or any parts to become disconnected and therefore the engine is always completely under control, and can never run away, no matter what the sudden variation in pressure or load.

The object of the governor is to preserve a certain determined speed with the smallest possible variation from constant speed as changes in the load occur. The cut-off must always be proportioned to the load. When no load is on steam is cut off very early in the stroke, and the

flying weights are at their extreme outer position; with a heavy load steam follows further, and the weights are nearer their inner position. Between these two limits any number of positions of the weights and corresponding angular positions of the cut-off eccentric may be had, and in each position as the steam is adapted to the load, the slightest increase or decrease in speed must make a change in the cut-off and bring the engine again to standard speed. In order that the governor may be very sensitive and instantly feel any variation of speed, it is necessary that the centrifugal force of the flying weights and the opposing centripetal force exerted by the dead weights and spring on the large bell crank be exactly balanced in every position they can possibly take; then any change of speed will cause the flying weights to instantly move in or out and be just as well balanced in their new position and as sensitive to any other variation in speed as they were before. This balancing of the centrifugal force, which is generated by the flying weights of the governor, is accomplished by the weights suspended from the horizontal arm of the large bell crank and the spring attached to the same arm. The governor is adjusted for whatever speed may be desired. To do this we hang enough weights on the horizontal arm so that, when the flying weights are at their inner position and revolving at their proper speed, the centrifugal force they exert is just balanced by the centripetal force of the hanging weights. Now, suppose an increase of speed occurs, then the centrifugal force of the weights increases beyond what we had before, and since we have opposed to it only the constant centripetal force exerted by the hanging weights, our balance no longer exists unless we bring in an additional centripetal force, which increases in the same proportion as the centrifugal force increases. This is accomplished by the spring attached to the horizontal arm. The tension exerted by the spring increases as the flying weights move outward, which is exactly what is required, and it will be seen that in all positions of the flying weights there is preserved an exact balance of the centrifugal and centripetal forces. The flying weights can therefore instantly move outward or inward when any change of speed occurs, and cut off the steam proportionately to the load. The arrangement of the weights and spring, as described, makes the governor extremely sensitive, and it also controls absolutely the speed of the engine, so that it can never run away, no matter to what amount may be the sudden change of load. We may start up with the throttle wide open and no load on (the flying weights are then held at their inner position by the hanging weights), and the moment the engine reaches the speed for which they are set and passes beyond that speed, the flying weights move outward and steam is cut off so as to maintain the standard speed. If a governor will take care of an engine with full throttle and no load except the ordinary fric-

tional one, it may safely be trusted to control the engine for any change that would occur when doing its regular work.

The mechanism of the governor is such as to permit of a very delicate adjustment. Referring to Fig. 6, it will be seen that the point of attachment of the spring may be shifted so as to get more or less leverage and extension of the spring, and therefore more or less centripetal force. There is a series of holes on the arm for this purpose. The governor weights and the tension of the spring are all calculated as closely as may be; then the final adjustment is made by attaching the spring at a greater or less distance from the fulcrum of the bell crank, and thus the balance between the opposing forces may be exactly determined, and the adjustment so accurately made that these forces increase and decrease in the same ratio.

There is a point to be noted in connection with this spring; the dead weights furnish a constant centripetal force to balance the centrifugal force of the weights when at their inner position. All the spring has to do is to furnish what is necessary to balance the increase of centrifugal force as the weights move out from the center, the initial tension is 0, its duty is light, it is never severely strained, and it has periods of rest, so that its elasticity does not become impaired. In this respect our governor differs from all those in which a spring is required to furnish all the necessary centripetal force. It is quite obvious that such a spring has a more severe and very injurious duty to perform, because it is always under tension, so that its elasticity soon becomes impaired and the governor does not act properly.

A comparison of the two methods of construction will show clearly the superiority of our governor and explains our very close governing under varying loads for which the Cummer Engine has achieved an excellent reputation.

As previously stated, the governor has its own shaft which is driven by a train of three small gears from the main shaft instead of placing the governor upon the main shaft itself. This arrangement secures for the governor and the general details of the engine very important advantages. In the first place, the work being light, the governor shaft need only be a fraction of the diameter of the main shaft, and we can reduce the size and friction of the eccentrics and eccentric sleeve, not less than eight to ten times what would be required if they were on the main shaft. Then, too, by having a separate shaft, all the valves may be in a direct line with the eccentrics, avoiding off-set rods and off-set rocker arms. Referring now to Fig. 7, it will be observed that, with a small shaft the points of attachment for the rods connecting the clamp collar on the eccentric sleeve with the governor weights may be brought much nearer the center of its shaft than would be

possible if the eccentric sleeve was placed on the main shaft. The advantage resulting from this is that, for a given angular movement of the eccentric, the eccentric on the main shaft has to move through an arc three to four times greater than the eccentric placed on a separate shaft of one-third to one-fourth the diameter of the main shaft, and consequently the weights in such a governor have to move in or out three to four times as great a distance as is necessary with a governor revolving upon its own smaller shaft, and is less sensitive, also, just in that proportion. The governor weights are placed at the most advantageous point for efficient governing; this is as near the rim as possible, and to this end also, the shape of the weights is such as to make the radius of gyration as great as possible.

The effectiveness and force of a governor weight varies in the ratio of the squares of the velocities, and as the velocity depends upon the radius of gyration it is easily seen why the weights should approach and recede from the center by only a small amount and this is permitted by the small movement required to operate the eccentric; and it follows also that, when the weights are at their inner position, and the engine is following ¾ to ⅞ of the stroke, the weights have moved inward so little, or from ⅓ to ¼ of what would be necessary if the governor was placed on the main shaft, that the governor has the valves and the engine as much under control as when in any other position. It is important that the governor weights be given such an adjusting movement that, when at their inner position, their force and value for governing shall not be impaired. From what has been said, it will be seen that the weights are always in an effective position and the governor acts equally well from 0 up to ⅞ of the stroke.

There are some other advantages connected with the construction of the Cummer governor. The governor case contains only the flying weights and their connections to the eccentric sleeve and thrust rod, the other essential parts are external to the governor and easily accessible, not only to examine their proper working but also to regulate the engine. In general a fixed speed is used, but if a different speed be required on one day from what is necessary on another, the change can be made without touching the governor itself at all; we have only to change the weights hanging from the large external bell crank, adding more weight to increase the speed and taking off weight to reduce the speed. Standard speed is restored quickly and certainly by hanging on the same weights which were there before. The gears composing the train which drives the governor shaft, are accurately cut and run without the least noise. They are contained in a case which constitutes their frame and covers them so as to prevent all chance of accident. The strain and wear of all parts of the governor are always in one direc-

tion, lost motion can never occur, and there is always prompt action. The various joints, bearings and all wearing surfaces are made large and are accurately fitted and easily reached for oiling and as they have such little movement and wear the governor may be relied on to last beyond the possible life of any engine. The governor may also be run for very long period without oiling.

The mode of attaching the governor weights through their rods and clamp collar to the eccentric sleeve, permits a ready adjustment of the eccentric. By loosening the bolts of the clamp collar, the sleeve and eccentric which is keyed to it, may be turned around to any desired position and again securely clamped, the adjustment being made to a nicety.

THE FRAME.

No part of an engine requires more careful designing than the frame because it has to receive all the working strains and resist them without springing, it must be so arranged that all the working parts connected with it may be properly located both for efficient action and for accessibility, it must be so designed that it can be well moulded without any undue strains in the casting itself, and must be moreover of graceful and pleasing outline, for the first thing to strike the eye and provoke criticism is the framework of the engine. We make a different style of frame suited to each class of engine that we build; we have endeavored to combine in them all the mechanical requirements together with beauty of outline, so as to produce a frame that is at once strong, rigid and well adapted to its work and also of graceful form.

It is very important to have a frame rigid, a very light frame may give sufficient strength but if there is springing or yielding under the strains from ordinary working, no amount of careful fitting is of any avail, because the frame does not keep its form so as to retain all the working parts in proper line.

We make all our frames very heavy and dispose the metal for flanges, ribs and braces in such a way as to secure the greatest stiffness with a given weight. There is an incidental advantage arising from heavy frames which is often lost sight of in designing machinery, and that is the fact that a heavy frame will absorb the shocks from working in such a way as to permit very little tremor, this conduces to long life and smooth successful working; a frame on the other hand which is not sufficiently stiff, and which trembles under a heavy strain besides giving an impression of insufficient strength is always at a disadvantage as compared with the former framing and is working its own destruction.

The general design for the frames of our engines is well shown by the full page engravings illustrating the several styles of engines made by us.

We will now describe in detail the frame for our engine *class* C, in which these principles will be found carried out and which may be taken as an example of what we aim to do in the designing of our engine frames. Following which will be found detailed descriptions and illustrations of the various parts of our engines, intended to show the main features of their design and construction, in the belief that their union of scientific and mechanical principles cannot fail to commend themselves to favorable consideration by those familiar with the requirements for successful engine building.

We have adopted for this class, the box girder form of frame as being the one which best answers our requirements for construction giving great rigidity, strength and stiffness without excessive weight, although the frame is made very heavy, To secure the greatest strength and stiffness, the girder is made very deep and is provided with heavy ribs and flanges, the main bearing is included in the same casting as that in which the guides are located, there is one continuous casting from the cylinder outward. This is an important feature and we wish to lay stress upon it ; where an engine is constructed in such a way that the part carrying the main bearing is bolted to the rest of the frame, there is always a sacrifice of rigidity to cheapness of manufacture. The frame must be stiff enough to stand up to its work and keep everything in line.

Fig 4 and Fig. 5 show the engine and frame in elevation and plan. The frame at its inner end is seen bolted to the cylinder which has an ample base resting upon the foundation to which it is fastened by bolts. At the outer end of the frame under the main bearing is another large base also securely bolted down. Some makers introduce a central bearing or support placed at the extremity of the guides, this idea probably originated for a frame too weak for its work. If any engine is properly designed the girder will be so proportioned that it is strong enough and stiff enough to be supported at each end only, letting the intermediate part take care of itself, which it should be abundantly able to do. The central support adds nothing to the alignment besides being wholly unnecessary and extremely unsightly. So persistantly has the necessity been urged for this central support by some other builders of girder frame engines, the frames of which were found to be deficient in rigidity at this point that occasionally a customer, not fully understanding the facts in the case, insists upon having this central support added to the engine ordered by him; we have uniformly resisted this when we could, but as we consider this to be an altogether

unimportant matter of detail we would not, of course, take the chances of losing a sale by refusing to furnish one should it be insisted upon by the purchaser.

THE MAIN GUIDES.

Fig. 10 shows a cross section through the frame at the guides fronting the cylinder, and exhibits not only the form of the girder but that also of the guides. These latter it will be observed are made flat and are removable, being attached by screws to the frame which is suitably planed to receive them. By having guides of this

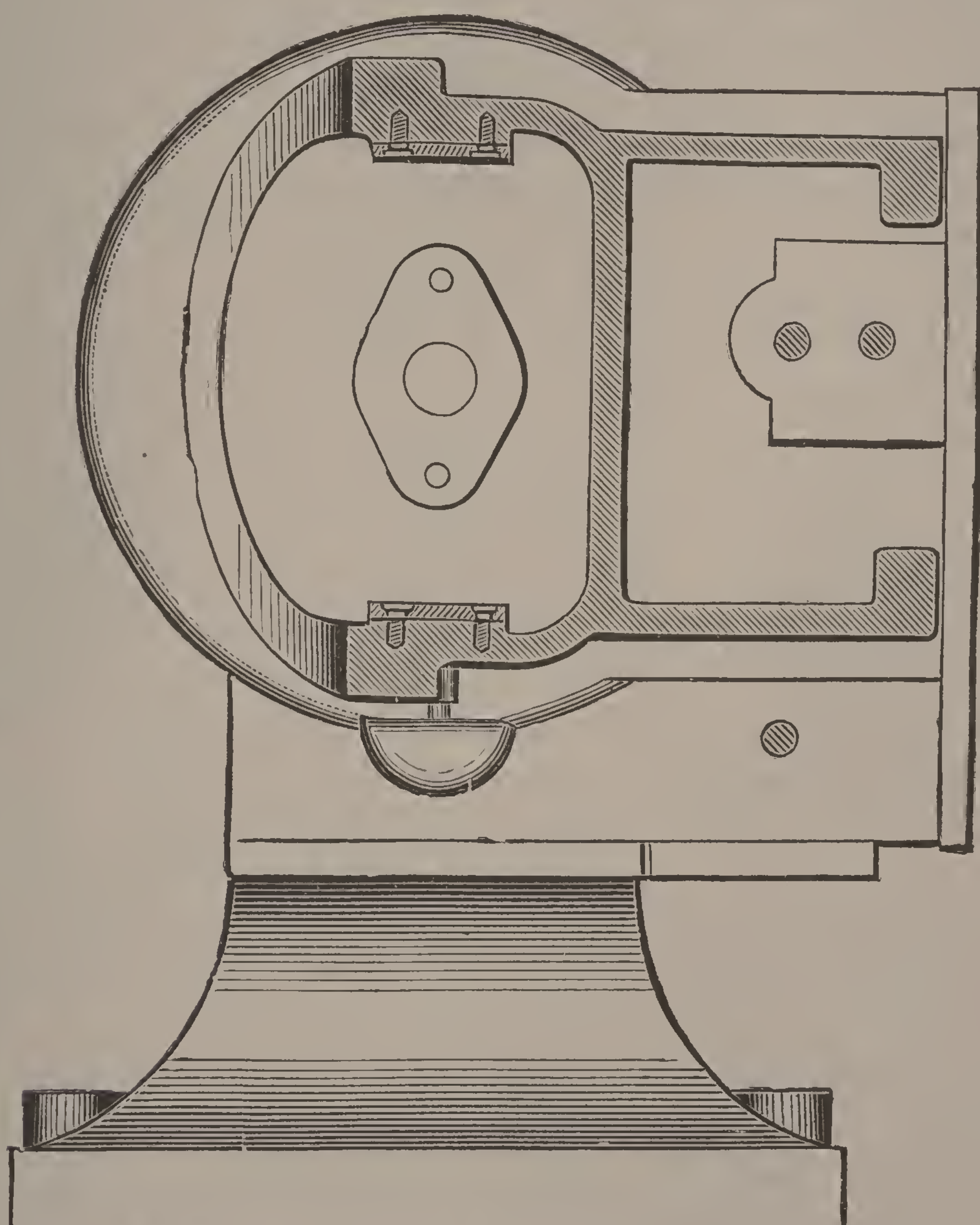


Fig. 10.

simple form and removable they may be easily and cheaply replaced whenever sufficiently worn to make a new set desirable. It is hardly necessary to enlarge upon the advantages which accrue from this important matter of detail as it is well-known that one of the most difficult and expensive items of repair about an engine is the refiting of the guides which may become necessary through long wear, by

reason of accident, or through neglect, if these guides are included in the same casting with the frame itself, for nothing short of taking the engine to pieces and having the guides planed over again will insure perfect alignment. The removeable guides are easily and cheaply made and can be put in place in a few minutes with ordinary tools and without skilled help.

THE CYLINDER AND VALVES.

In order to more clearly understand the construction of the cylinder we will give a description of the valves employed, prefacing it by a brief consideration of what kind of duty they are required to perform. The common D valve though apparently very simple, is somewhat complicated in its action. It has to admit steam to either end of the cylinder, to act as a cut-off within certain limits, and also to exhaust the steam. It is found that when much expansion is used that a derangement of the exhaust takes place; if we cut steam off early, the exhaust also is closed early, whereas it is desirable to keep the exhaust open until late in the stroke. The two requirements being so different, cannot be fulfilled by a single valve without impairing the action of each in the endeavor to harmonize both. The idea then occurs of using a separate steam and exhaust valve whose actions are independent, in order that each may be permitted to work to the best advantage. This arrangement we have adopted for our engines. There is a steam valve with its steam chest, ports and passage, and a separate exhaust valve with its own chest, ports and exhaust passage. We use flat valves and for expansion have a small cut-off valve sliding upon the back of the main steam valve. This cut-off has its own eccentric, which is connected with the governor so as to cut off proportionally to the load. The main valve and the exhaust valve are operated from another eccentric and have each the same travel. A small amount of lap is given the exhaust valve just enough to cover the ports, while the steam valve whose ports are smaller, has enough lap added to compensate for what lead is given the exhaust valve. The lap on the main valve, however, is not enough to cause a cut-off within the limits assigned the governor. For the form of valve, both steam and exhaust, we have adopted that known as the "gridiron," being a simple flat plate with narrow openings through it. This form of valve gives a large area of opening for a given angular advance of the eccentric and for a port area of given length, this enables us to reduce the throw of the valve in proportion to the number of openings through it. Thus in a valve with four openings whose combined area equals that of a single port such as is controlled by the ordinary valve, we will need only one-fourth as much travel; there is required ordinarily a movement of only ½ to ¾ of an inch, and we reduce the friction of the valves in proportion, by this

means we can also use much smaller eccentrics than would be possible with an ordinary engine, since the travel required of the valve is so much less, but a still further reduction in the diameter of the eccentric occurs in consequence of using a separate governor shaft, which may be made of smaller diameter, instead of placing the eccentrics on the main shaft as is usually done. This arrangement which permits the use of eccentrics several times smaller than the usual mode, reduces the friction of the straps in the same proportion.

THE VALVES.

Fig. 11 shows the main valve in plan and section,and Fig.12 the cut-off plan and section. There are three openings in the main valve, corresponding to the three ports of the valve seat. The cut-off valve admits steam from one of its outside edges and from corresponding inner edges of its two openings. Fig.13 shows the exhaust valve which has three openings but as one of the outside edges admits steam also, there are four ports in

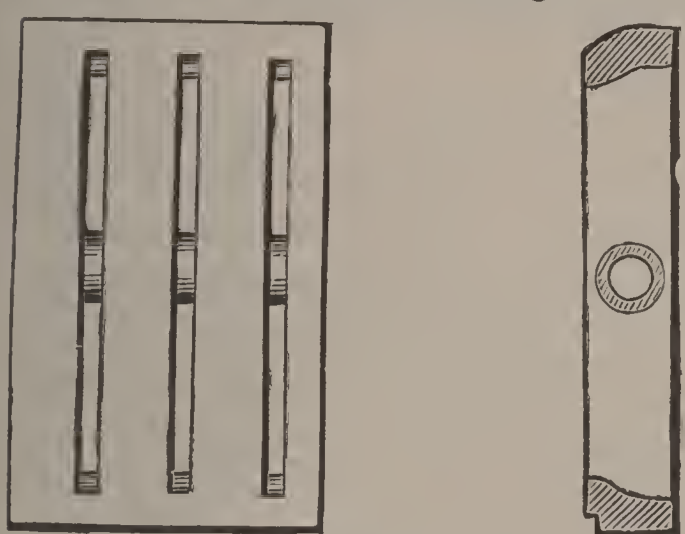


Fig. 11.

Fig. 12.

its valve seat. It is very important to have in both steam and exhaust valves a large opening at the moment they are intended to act, in the one case we wish to get steam into the cylinder as quickly as possible without any wire drawing, and in the other to have a full opening for exhaust so as to reduce back pressure as much as possible. With an ordinary valve and a single port this can only be accomplished by giving excessive lead with its attendent evil of too early release, but in valves of the construction just shown having several openings a small amount of lead suffices to make a large area of opening and we secure the desired result without sacrificing any other advantages. Each end of our cylinder is fitted with its own steam and exhaust valves, this is done to make the steam passages short and direct, thus reducing clearance, and also to admit of a short travel for the valves. We have thus secured all the points desired, there is the steam valve with its cut-off and another separate valve for exhaust; they can be adjusted independently so as to give each steam or exhaust valve just the lead or compression that may be re-

quired for successful running. In order to the better attain this end we make the valves for each end of the cylinder so that they may be adjusted independently to secure proper action of the steam at each

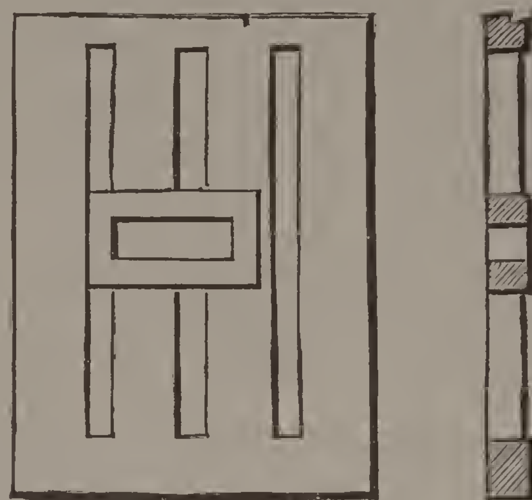


Fig. 13.

end. We have reduced clearance to a small amount and lessened the friction and travel of the valves several times below that attained by valves as ordinarily constructed.

We do not advocate a balanced valve because, although they are undoubtedly good when first made the fact is they do not long remain in proper condition, they cannot be kept tight. Our valves are not provided with special means of balancing, but the power required to move them is not great owing to their short travel and small size and from having several openings still further reducing the area exposed to steam pressure. The exhaust valves have very little friction; that valve which is exhausting is free from pressure and the other one has full pressure upon it only up to the point of cut-off and is then relieved as the steam expands and its pressure falls.

THE CYLINDER.

Such being our valve construction the cylinder is designed to meet these requirements. Fig. 14 shows an elevation, Fig. 15 a longitudinal section and Fig. 16 a cross section of the cylinder in which appear the general features and arrangement of parts. The elevation shows in part-section the steam passage with a short length of pipe, and below it the exhaust passage. On either side are the steam and exhaust chests; that on the right has the cover removed showing steam and exhaust valves. The valve stems are also shown, they pass through the steam and exhaust passages and connect the valves at one end with those at the other. Fig. 17 is a cross section through the cylinder in a plane between the centre and the nearest end, it shows the steam pipe and passage, and valve stems, below is the exhaust steam passage whose position is so isolated that very little loss of heat can occur by passing off with exhaust steam. Fig. 16 is a cross section through the cylinder and its base; the plane of sec-

tion being through the valve chest, shows the steam valve with its cut-off and underneath them the exhaust valve sliding in a horizontal plane. We make the exhaust valve seat removeable for greater convenience of construction and to allow for refacing. Below the exhaust valve is seen

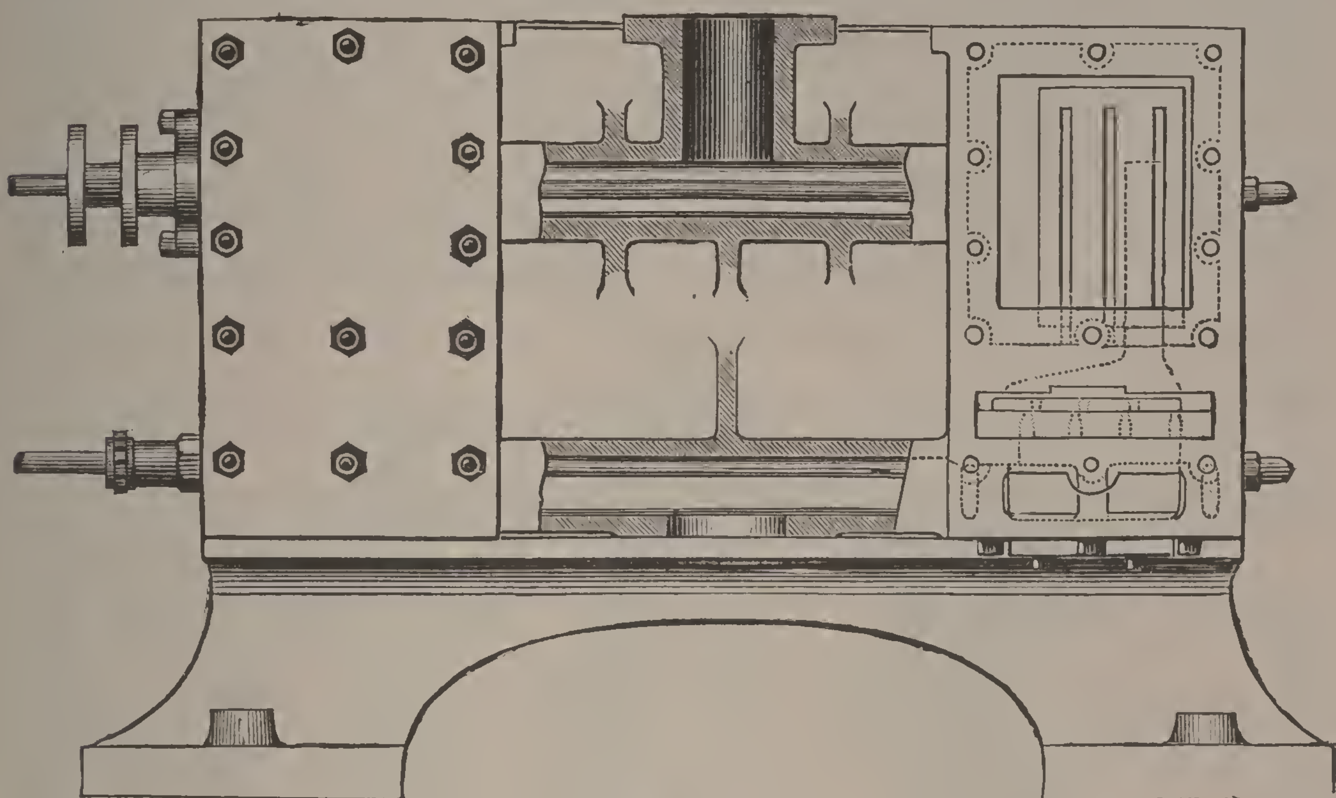


Fig. 14.

the exhaust passage and there is also shown the means by which the valve is moved; this consists of a clamp collar which fits the valve stem and is provided with a prong coming up through a slot in the valve seat and attached to the valve. One-half of the horizontal section (Fig. 15)

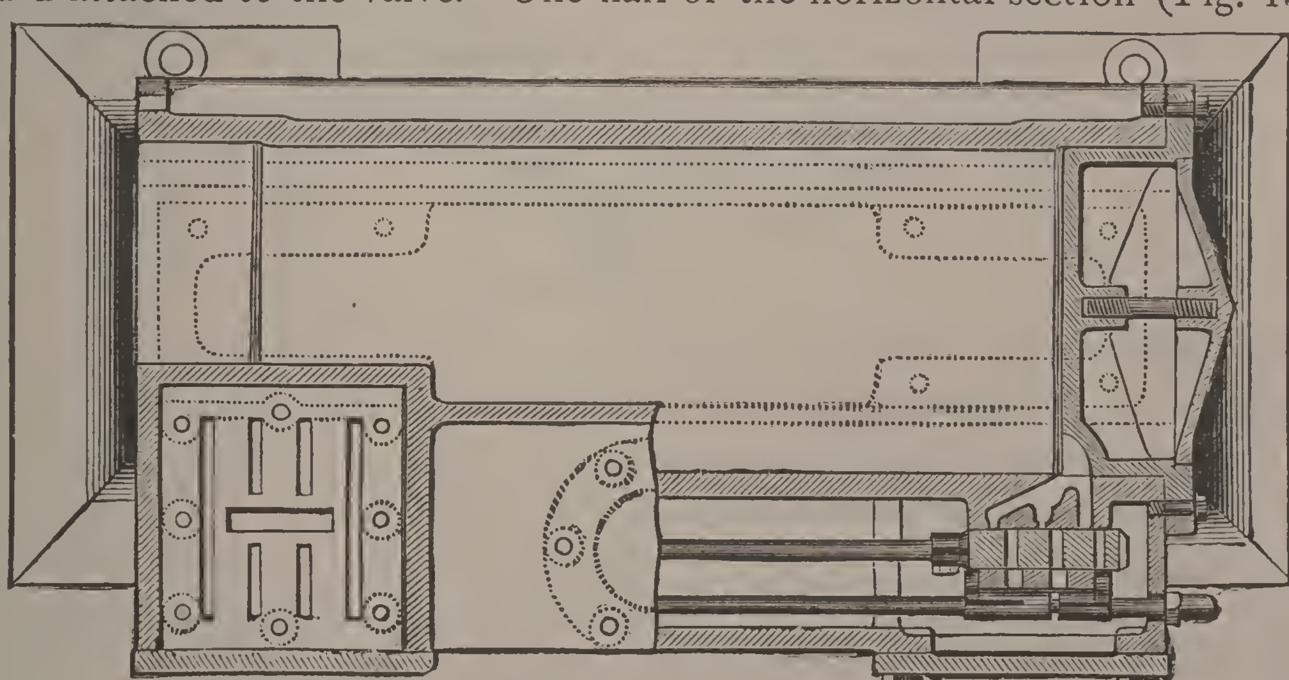


Fig. 15.

shows the main and cut-off valves for that end with its steam passage. The other half is sectioned to show the exhaust valve seat. In this figure and also in the cross section (Fig. 16) we see how steam enters the cylinder through a three ported seat afterwards uniting to form but one opening into the cylinder, and how steam is exhausted through the lower part of

this same port which then by one large opening communicates with the chest in which the exhaust valve is located. There is no real connection between the two, however, each valve within its own chest controls its own ports, and live steam cannot enter the exhaust. In Fig. 15 there is a sectional view of one cylinder head, this is inclosed by a conical shaped cover fitting air tight, its surface is polished so as to look well and prevent radiation, while the air in the interior space acts as an excellent non-conductor of heat. The head for the opposite end is formed by the inner end of the frame which fits into the cylinder and is bolted to its flange.. The cylinder itself is covered with non-conducting material and the whole inclosed by a neat cast iron lagging.

We make our cylinders of carefully selected iron so as to produce a metal of great strength, hardness and close grain. There is a generous

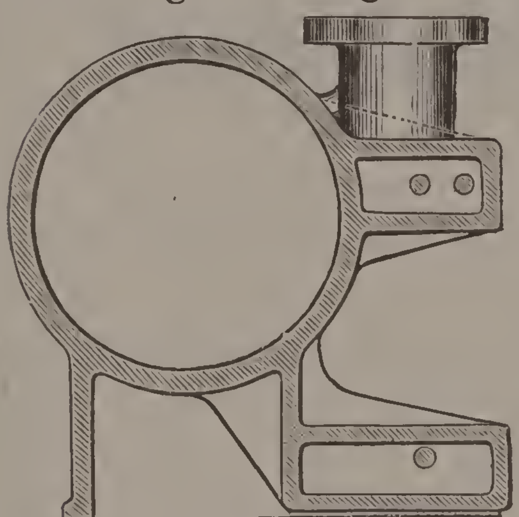


Fig. 17.

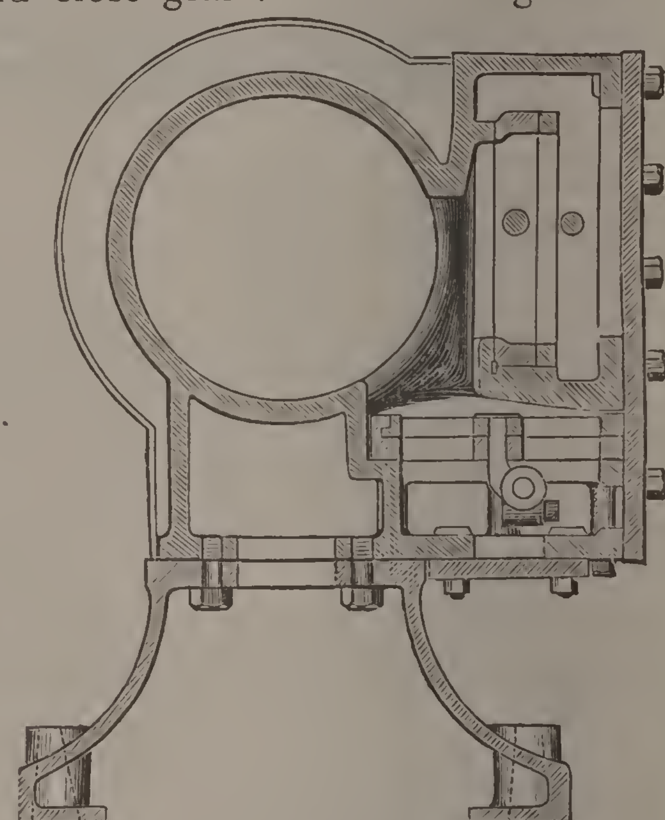


Fig. 16.

allowance of weight so as to be stiff enough to retain its shape and not spring or distort under any strain and also to allow for reboring. The valves are carefully scraped to an accurate bearing and being flat are easily fitted and remain tight for a long time. Both steam and exhaust valves have a constant travel under all conditions and this conduces to equal wear, while from the simple construction whenever repairs are needed they may be easily made in an ordinary shop with ordinary tools. This is a very desirable point. The valve and valve stems as well as the eccentrics are provided with means of adjustment so that the desired amount of lead and cut-off may be given the steam valves, and the exhaust valves set for the desired release and degree of compression. Each valve can be adjusted independently of the others so as to act in the most efficient manner. In the different cuts it is shown how readily accessible all the parts are both for adjustment and repairs.

THE MAIN BEARING.

As already mentioned the main bearing forms part of the same casting as the girder frame, Fig. 18 is a section through the bearing and shows clearly each part for large engines. The bottom box is of cast iron filled with a very hard and tough anti-friction metal, it is planed to fit a place of corresponding shape in the frame and follows the alignment of the shaft, and when worn out the bearing can easily be repaired or a new box slipped into place. For small engines the anti-friction lining is included in the casting of the main bearing as shown in Fig. 19

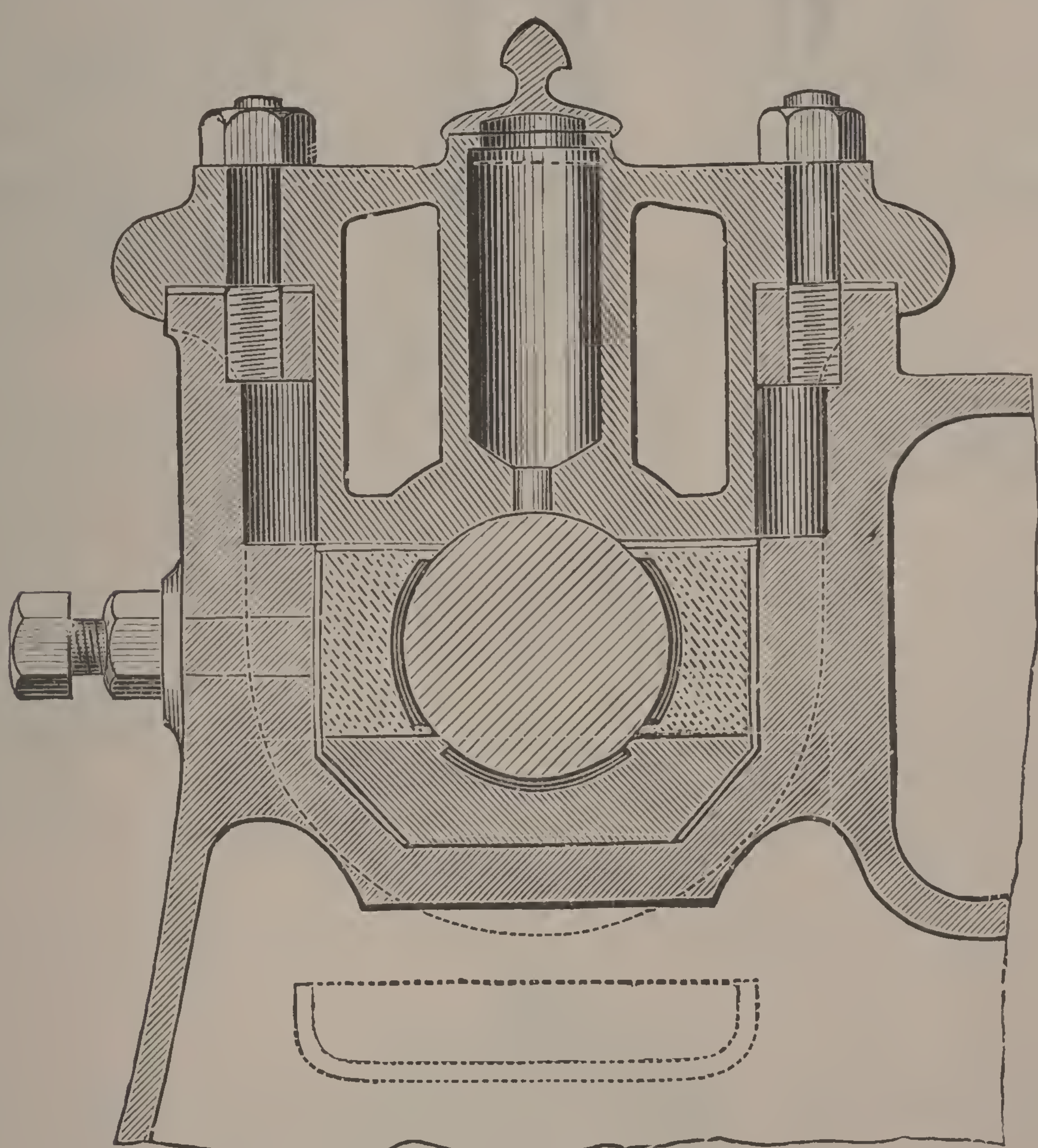


Fig. 18.

Side gibs are provided to take up wear, these are adjusted by set screws on one side, and on the other side by thin strips of metal which extend over the whole flat surface, these prevent the gibs from getting out of square which they are liable to do when set screws are used on both sides. The boxes are lined with antifriction metal and then having the cap bolted on everything is bored out together making a true hole. The length of the bearing is twice its diameter and the diameter approximates one-half the diameter of the cylinder. A large central cavity is shown in the cap, this is to be filled with a lubricating compound, and a

piece of heavy copper wire passes down through the oil hole until it touches the shaft. The friction develops a moderate temperature which being conducted upward by the copper wire, melts the compound and lubricates the shaft. This device works well, but any other of the ordinary modes may be used if so preferred. A drip pan shown by dotted lines, catches any surplus oil falling from the journal and secures cleanliness.

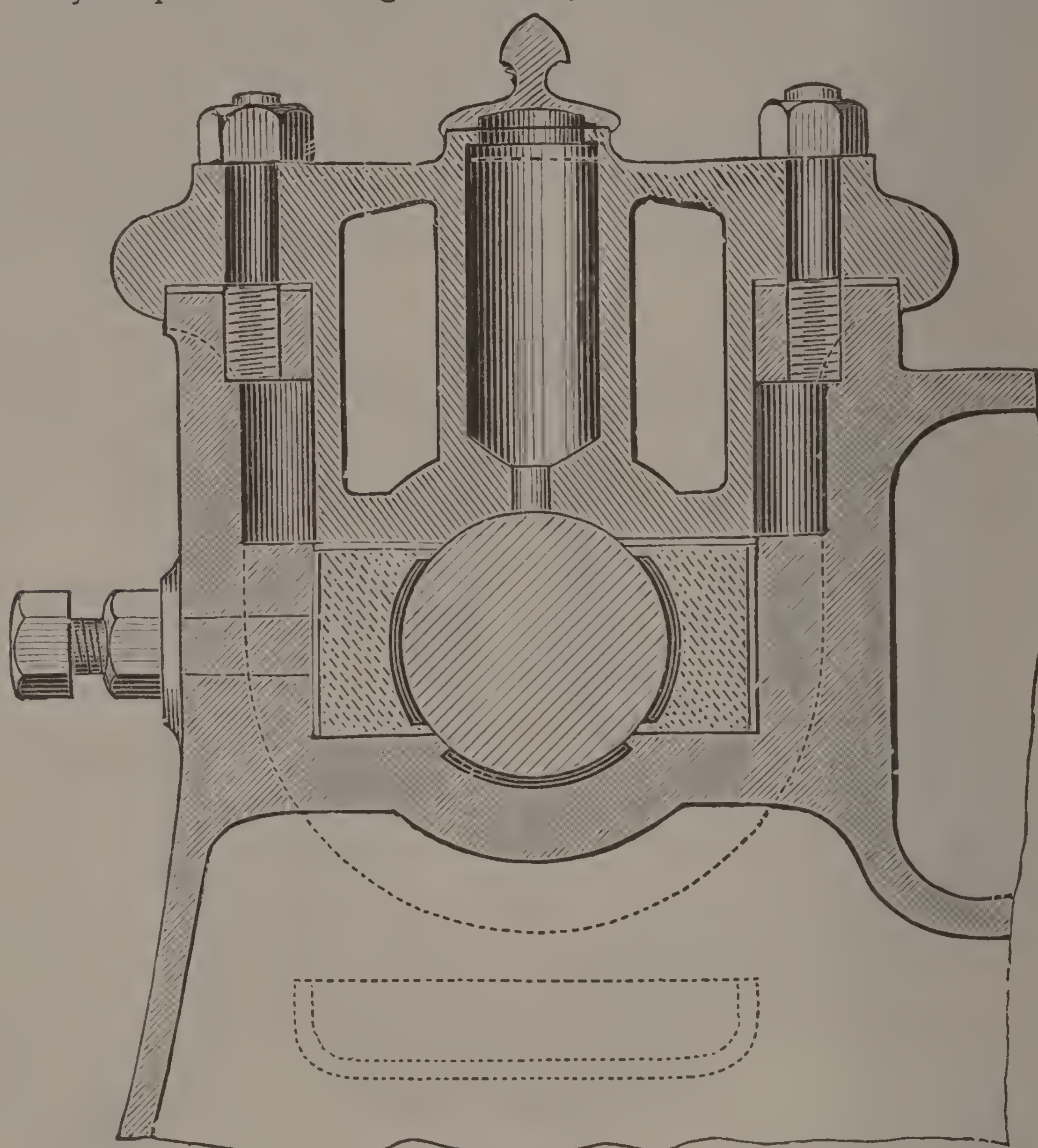


Fig. 19.

CROSS HEAD.

The cross head is of neat and strong design and suited to the work it has to perform. The sliding surfaces are made flat and planed to fit the guides, their form admits of easy and exact fitting and are of ample surface for the constant pressures coming upon them. When it is necessary to make the final adjustment, or afterwards, in order to take up anywear, means have been provided to explain which, reference is had to Fig. 20 where the gibs are shown in elevation and section. The gibs are held in position in both directions by the hooked ends and dowel-pins as shown they (the gibs) are adjusted outwardly by means of the four taper keys, two to each gib, which appear in both views. These keys are square in section and furnish a solid backing for the gibs. The

figure gives at once a section of the cross head and gibs showing one of the keys and its nut, the connecting rod pin, and also one of the dowel-pins for holding the gib in position sideways. The piston rod is fitted into the boss seen at the left side of the cross head and securely held by means of a cotter. The connecting rod pin is placed at the centre of the cross head, this is the most favorable position it can have because it distributes the pressures from the connecting rod, evenly over the

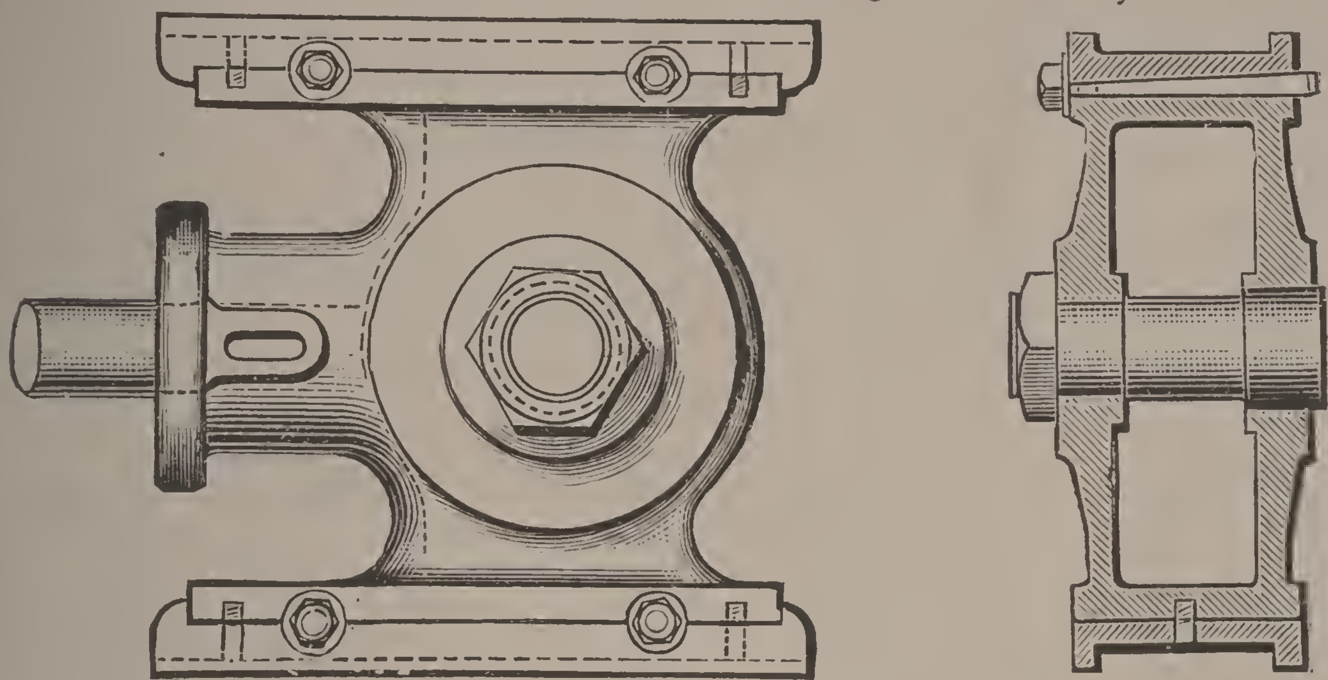


Fig. 20.

whole wearing surface. There is no unequal wear from excessive pressure at one point or side strain upon the piston rod as is always the case where the point of attachment is placed beyond the centre, or overhangs the gibs. There is no harm to result if the boss to which the piston rod is secured overhangs, because the strains are never sideways, but it is a serious defect to place the connecting rod pin outside of the centre as is evident upon considering the strains which have to be met.

THE OUTBOARD BEARING.

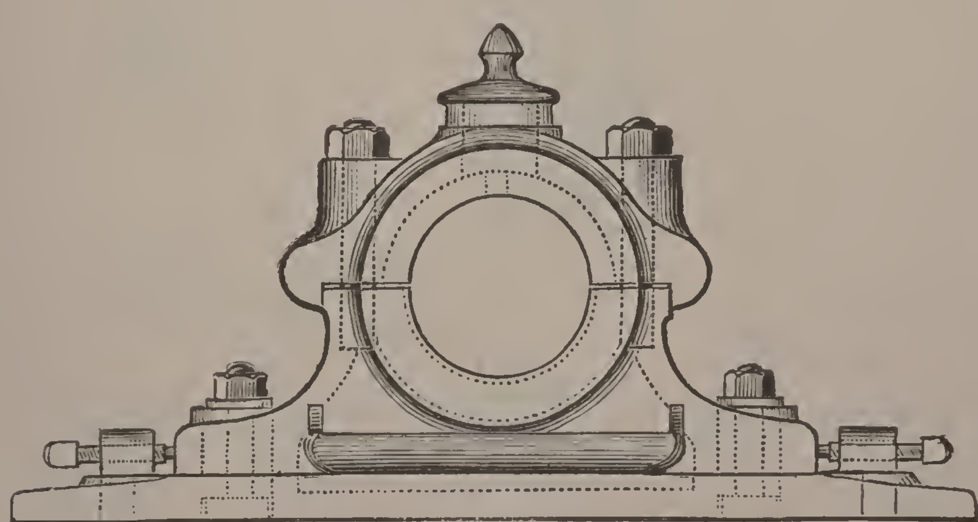


Fig. 21.

Besides the main bearing, there is an outboard bearing shown in Fig. 21; It is provided with a heavy cast iron sole plate which rests upon the foundation and is securely bolted thereto. The bearing proper rests on

this plate, the surfaces in contact being planed. Adjustment may be readily made in either direction, without removing the shaft. The bearing and sole plate are bolted together and all are firmly fastened to the foundation. The bearing is lined with anti-friction metal and then bored out. Similar means of oiling are used as with the main bearing and there is a drip pan at each side to preserve cleanliness by catching any waste oil which may escape at the ends instead of allowing it to drip down on the foundation walls.

THE CONNECTING ROD.

As already referred to in a preceeding paragraph the connecting rods on our engines are made so that the distance from centre to centre always remains the same after adjusting for lost motion ; the only chance for variation will be unequal wear of the boxes. That end of the rod

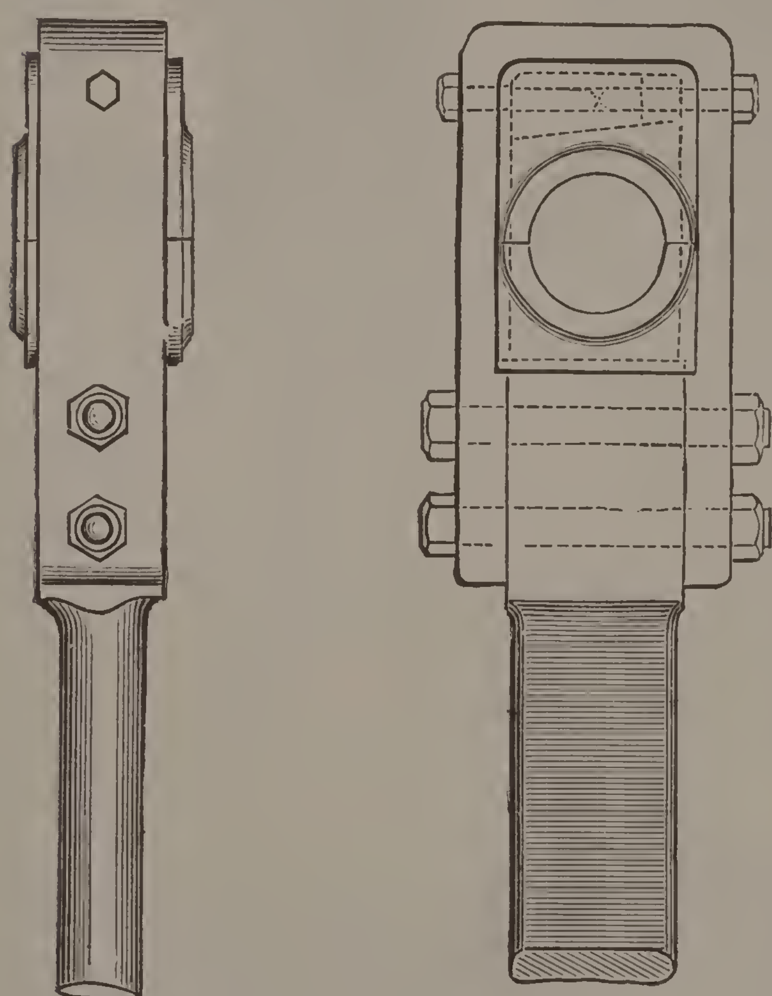


Fig. 23.

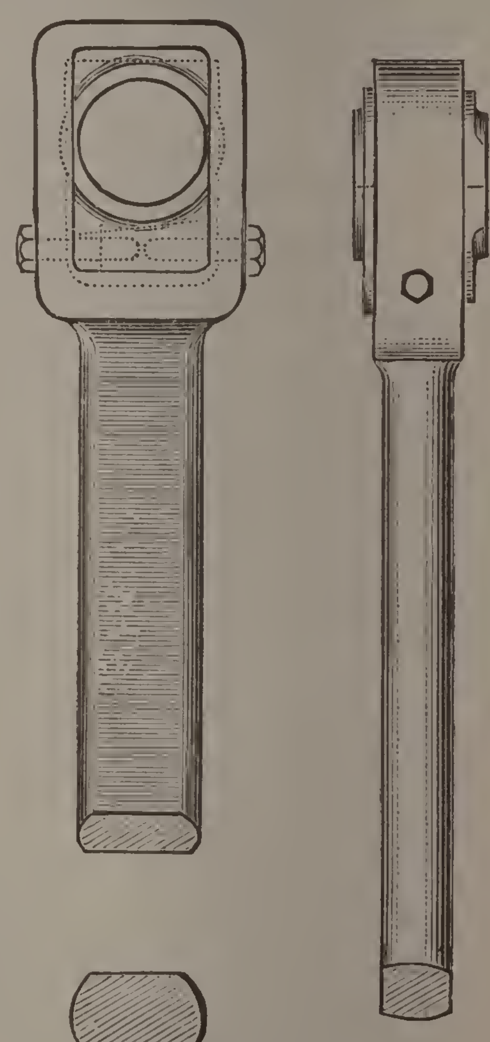


Fig. 22.

attached to the cross head is forged solid, no strap, gib or key being used; the adjustment for wear being made by altering a sliding wedge which fits against the box and the slotted portion of the rod see Fig. 22.

The crank pin end of the connecting rod is fitted with a forged strap held in place by two through-going bolts and nuts, making it almost equivalent to a solid end.

The straps and the two brasses are somewhat less in distance than that portion of the rod through which the bolts pass, the object of which is to permit refacing the inside of the strap in after years when, by reason of long wear under heavy loads, there might be so much lost motion between the brasses and the strap as to render such re-fitting desirable or necessary. The allowance we make is such that, when the strap is re-fitted the brasses can be placed in position without having to distort the strap by springing it open to get the new brasses in. The adjustment for wear is the same as that described for the cross head end of the rod, see Fig. 22.

The rod and strap are both forged under heavy hammers from selected scrap iron, and are fitted and finished in the best manner. Self acting lubricators are supplied from the best designs we can get. The boxes are made from new copper and tin in such proportions as will give the best results under the heaviest loads.

THE CRANK.

We use a strongly made cast iron crank which is carefully fitted to shaft and forced on by pressure. The crank pin is made of mild steel and carefully proportioned. Besides an oil cup on the connecting rod, which is the ordinary method used to oil a crank pin, we employ a device shown

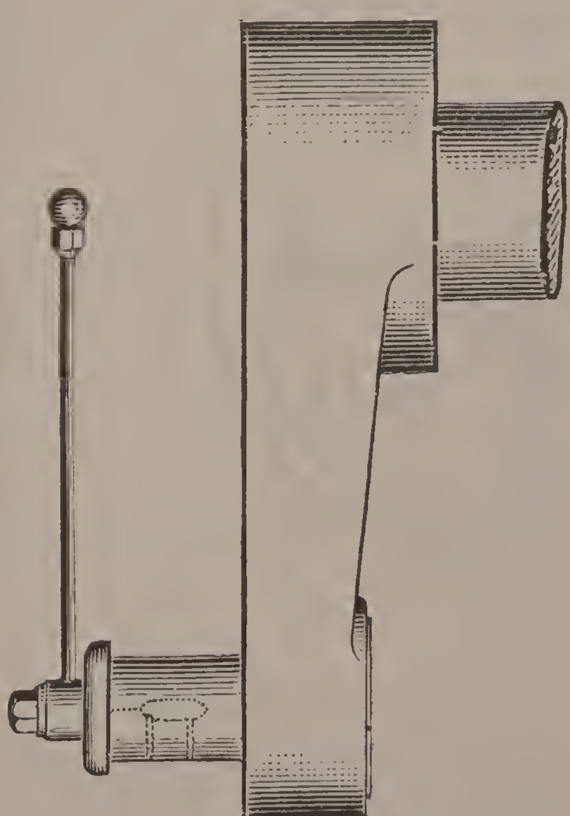


Fig. 24.

in Fig. 24 This arrangement consists of a tube, connected to the crank pin at one end, and at the other carrying a ball whose centre is in a line with that of the shaft and it therefore remains practically at rest simply turning around without moving out of centre. Holes are drilled in the

crank pin as shown by dotted lines and oil may be introduced into the ball when running and is thrown outward by centrifugal force so as to reach the surface of the crank pin. This simple device is an important addition for the ordinary method of oiling may give out and as the crank pin of an engine is one of those points requiring always to be kept well lubricated, any means of accomplishing it with certainty and without the necessity of stopping is very desirable.

THE PISTON.

The piston is made large enough to give ample wearing surface and with sufficient weight and careful distribution of metal to secure strength, no extra weight being given for any other purpose. The piston consists of three parts, the piston proper, to which is fitted the tapered end of the piston rod secured by a cotter, the chunk ring and the follower. These parts are all clearly shown in the section. The plan exhibits the piston with its follower removed but showing in section the four bolts which holds it in place when connected. It will be noticed that the

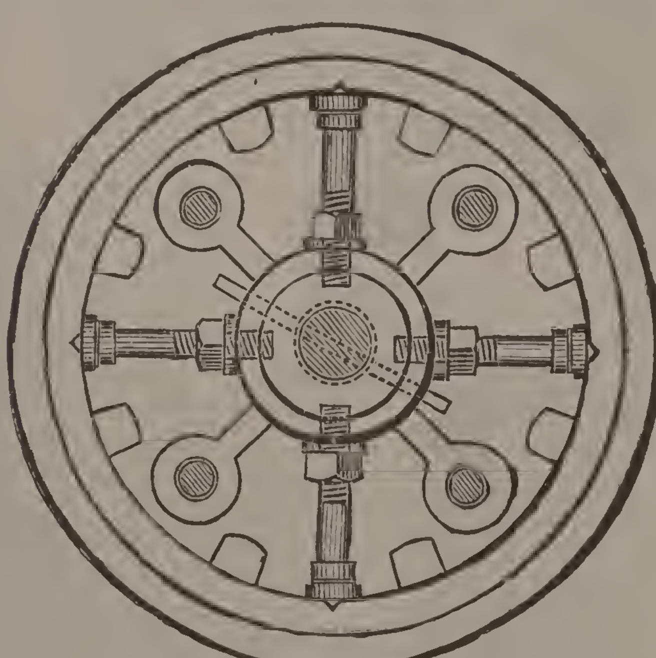


Fig. 25.

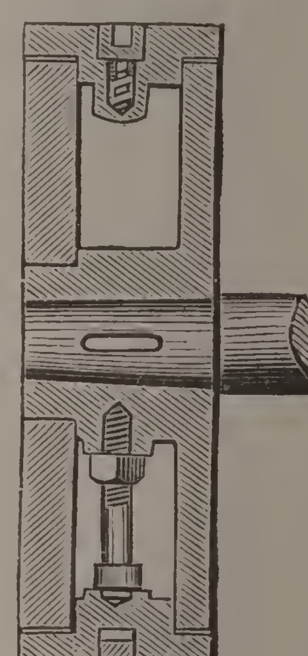


Fig. 26.

piston itself and also the follower are made considerably smaller than the cylinder and that the chunk ring is external to these and forms the bearing surface. The chunk ring is turned up so as to be an accurate fit and is then adjusted so as to be perfectly central by means of four stud bolts, which appear in the plan and section; their outer ends have a conical point which bears against the chunk ring while the other ends are tapped into the boss of the piston and are provided with jam nuts. The centre of the chunk ring is grooved to receive the cast iron piston ring which is pressed outwards by several small spiral springs spaced around the cir-

cumference. The positions of these springs appear in the plan and one of them is shown in the section. An additional packing is provided by turning two small grooves in the chunk ring on either side of the central piston ring. The advantage of using a chunk ring is that, we can make a very exact fit, and by using the central adjustment secure perfect alignment, and we obtain a greater wearing surface for the same thickness of piston because the chunk ring is the same width as the piston itself and bears over its whole surface, whereas in the ordinary form a part of the piston, and the follower also, are turned down below size and do not bear at all. By this arrangement also, whenever after long wear it becomes necessary to rebore the cylinder we have only to turn up a new chunk ring instead of fitting up a whole new piston.

INTERCHANGEABLE PARTS.

By our system of manufacture all the parts of our engines are made exactly alike for any given size, this enables us to furnish duplicate parts at once as we shall always have in stock such portions of engines as are liable to wear out, or are most likely to be injured in case of accident. This feature will, we are sure, commend itself to all business men as it enables purchasers of our engines to have any part of an engine shipped at once upon mention of the part wanted by telegraph or otherwise.

MAIN SHAFT.

The main shaft is made of carefully selected scrap iron forged under heavy hammers to ensure thorough working. We give unusually large dimensions to the shaft itself, and make the bearings long and of large diameter. Owing to our arrangement of governor and eccentrics on a separate shaft, any repairs or adjustment to these parts may be made without disturbance to the main shaft, and there are also other advantages which have been already referred to elsewhere.

FLY-BAND WHEELS.

Our belts for each engine have been calculated to transmit the power developed at the given rate of revolution and a certain surface velocity of belt, a liberal factor of safety being provided in the formula employed. The width of belt and its velocity determines, in each case, what size band wheel is to be used, and since a band wheel has generally to fulfill also the office of a fly wheel, its rim must have the weight necessary for good regulation, just as if it were a fly wheel

rim, and the weight must be calculated with equal care. Our formula for fly wheels and band wheels is such as to give ample weight of rim in all cases; we also make the arms and boss of large dimensions, and special care has been taken in the design to provide against initial strains in the casting itself, which occur when the metal has not been properly distributed. The proportions adopted to secure a strong, reliable casting are the result of a large number of experiments, and have always given full satisfaction.

Band wheels proper for each size of engine have been calculated, and will be found in the various tables; in general, it is best to adhere to these sizes, but if a different size be required to secure a certain velocity of line shaft, we will, upon being informed of the requirements, furnish the proper sized wheel. It sometimes occurs that a fly wheel or fly-band wheel must be made of a smaller diameter than is given in the tables, this will, in almost all cases, necessitate a very considerable increase in the total weight of the wheel in order to maintain the same momentum which we have allowed for the wheel of larger diameter; in all such cases we shall make an extra charge for the difference in weight between the two wheels.

FLY WHEEL.

Fly-Band wheels will be used with most engines, but in cases where an engine is coupled directly to the line shaft, a fly wheel is employed instead of a band wheel. We will furnish either wheel for the same price, one being considered the equivalent of the other. Similar care is exercised in their design and construction as with our band wheels; they will be found strong and well-proportioned and of great weight of rim, so as to regulate closely as pointed out under the Theory of Fly Wheels.

COUPLINGS.

The couplings to connect the main shaft and line shaft of an engine employing a fly wheel, will be furnished by us when so desired, or we will make only the half coupling for the main shaft; in either case the price of couplings is extra. Where couplings are used, the outboard bearing must be located far enough inward to give the required space for a coupling to be attached.

FOUNDATIONS.

It is important for all classes of engines to have a solid, immovable foundation so that, when once set up, the engine may keep all its working parts in line, so as to work smoothly and without heating, and that there may be no springing or tremor under such strains as occur from the development and transmission of power.

Especially do automatic cut-off engines require good foundations, both to support the extra heavy weight and to resist the strains resulting from high speed and expansive working. It is always poor policy to economize in foundations, and in the case of an automatic engine, one may largely sacrifice the advantages which have been costly to secure. We have been at pains to give a proper form to the engine frame, to have large surfaces to support the weight, and to give all the working parts correct proportions and accurate fitting. To secure the advantages resulting from careful design and good workmanship it is then highly important to supplement them by a good foundation which gives the requisite rigidity and preserves the correct alignment required for smooth, efficient working.

A foundation built up of irregular pieces of stone is thought by some to be sufficiently good for the purpose, and in localities where stone is plentiful, there is a tendency to build them in this way. But in reality such rubble masonry makes a very bad foundation, because the irregular pieces touch each other only in a few points, instead of having a solid bearing over the whole surface, and the interstices are filled in and wedged up with smaller pieces of stone and mortar, so that any strain coming upon the mass, cannot meet a solid resistance, but throws the stones out of place. There is then a kind of re-adjustment of the various pieces to each other, but they only remain in this position until they meet with another strain. Such foundations are thus continually liable to change their form and even jar to pieces, and they never do what is required of them, which is to furnish an immovable, solid base. Cut stone foundations are not liable to these objections, but they are very expensive, and it is, besides, unnecessary to employ them.

There is no better foundation to be made than one constructed of good hard brick, well laid with thin joints of good cement. Foundations such as this are easily and cheaply built, and when sufficiently set, are firm and unyielding, being fitted to support all the weight and resist without change of form all the strains coming upon them.

When bricks are not obtainable, a stone foundation must be employed, and we recommend the following method of construction, which does not have the objectionable features of rubble masonry. Take good-sized dimension stones and lay up a course dry; point the outside

stones with mortar or cement and fill up the interstices between the stones in the interior space with gravel, or small pieces of broken stone, but preferably gravel where this may be had; then pour in grout—which is a thin mortar made either of cement alone, or cement mixed with one to one and a half parts of sand, using enough water to allow the grout to flow easily, so as to penetrate the whole mass of stone-work, and then, having levelled off the surface, commence laying another course of dry stone. Proceed in this way until the foundation is of the required height. When well set this makes one solid block of masonry and is an excellent foundation when brick cannot be had; it is preferable to brick where the soil is wet, and it is good practice also to use concrete for that part of a foundation which extends below the water line.

For that part of the foundation above the ground and upon which the cylinder, main bearing, and outboard bearing directly rest, we use large solid blocks of cut stone. This is done in order to better distribute the weight and working strains over a large surface, which could not be done so well if brick alone were used for the whole foundation, because the flanges of the frame do not cover so large an area as the stones. We specify in all cases where the soil is dry, a plain brick foundation constructed as above and provided with solid blocks of cut stone beneath the cylinder, main bearing and outboard bearing. The foundation stones are solidly set in cement on the brick work, and after lining up the engine the spaces between the engine frame and stones are filled with melted sulphur.

The nature of the soil or rock upon which a foundation is to rest, and the weight of the engine itself, will determine the character of the foundation and the dimensions to be given in any particular case. Moreover, the plans have to be made with reference to the arrangement of engine, driving belt, and other special requirements for each case. We desire always to be made acquainted with all necessary particulars, and will then furnish a full set of foundation drawings suited to the requirements. When so requested, these drawings may be sent in advance of the shipment of the engine. Foundation bolts and anchor plates are furnished with each engine. In order to accommodate slight variations in the lengths of frames, the bolt holes are made larger than the diameter of bolts.

CONDENSERS.

A good condenser will increase the economical power of an engine from 20 to 40 per cent., or for the same power effect a corresponding saving in the amount of steam used and fuel consumed. With an engine of any considerable size, a condenser may always be employed with economical advantage or we can when desirable increase the power of an engine of given size without adding anything to the initial steam pressure or boiler capacity. Condensers owe their efficiency to the

fact, that they create a partial vacuum on the exhaust side of the piston and thus reduce back pressure in proportion to the perfection of the vacuum. Atmospheric pressure, such as non-condensing engines work against, amounts to 14.7 lbs. per square inch; from 11 to 13 lbs. of this may be removed by means of a condenser and is just so much added to the mean effective pressure, without any additional cost, except for power required to operate the air pump which gives the injection and removes the condensed steam and injection water, and, as elsewhere explained, the steam necessary to develop this power need not be lost when we employ heaters. Since a condenser will thus add so largely to the power and economy of an engine with but slight additional outlay, we recommend its use wherever a sufficient supply of good water can be obtained for injection.

It is our practice to employ, whenever the conditions will warrant it, an independent condensing apparatus; because, the vacuum is had at starting and may always be maintained regardless of the speed of the engine or varying temperatures of the injection water; we can use a smaller air pump and do not have to operate at all times a larger pump than necessary in order to provide for emergencies; and, the power required to operate the pump does not act in any way to disturb the working of the main engine.

The amount of injection water required is from 20 to 25 times the quantity fed to the boilers. Water discharged from the condenser has a temperature of 100° to 115° F. A portion of this water may be fed to the boiler, but by far the greater part has to run to waste. We may remark in passing, that this is a serious and at present unavoidable source of loss, which is, to a greater or less degree common to all steam engines. In round numbers if 1100 heat units are contained in one pound of steam entering the cylinder, from 900 to 1000 of these units are carried off by the exhaust steam and imparted to the condensing water. This explains why we can only realize a small percentage of the power contained in each pound of coal, only about 4 per cent. to 16 per cent. can be counted upon and only about 29 per cent. is possible supposing steam of 90 lbs. to be expanded down to the line of perfect vacuum; the remaining heat is necessarily lost because there is no means by which any further expansion and resulting work can be secured. But while the percentage of power obtained is low compared with the power which could be realized with perfect mechanism extracting all the heat, yet compared with the amount of heat which is possible to utilize, it may be shown that some of the best types of engines yield about 50 per cent. of the highest efficiency; and, future improvements may be expected to increase this figure which is still so far below what may be considered attainable.

INDEPENDENT CONDENSING APPARATUS.

This condenser which we use with our engines, is of simple construction and effective in operation, it is compact and self-contained requiring no special or expensive foundation. No difficulty will be experienced in applying it to any non-condensing engine, the attachments are all easily made while the small amount of space occupied permits it in most cases to be placed beside the main engine and where an engine room is small, the condenser may be placed below the floor in such position as to be easily accessible. The condensing apparatus it will be seen is a modification of a Deane pump, only instead of pumping water alone, we pump both water and air, together with any vapor that may be in the condenser.

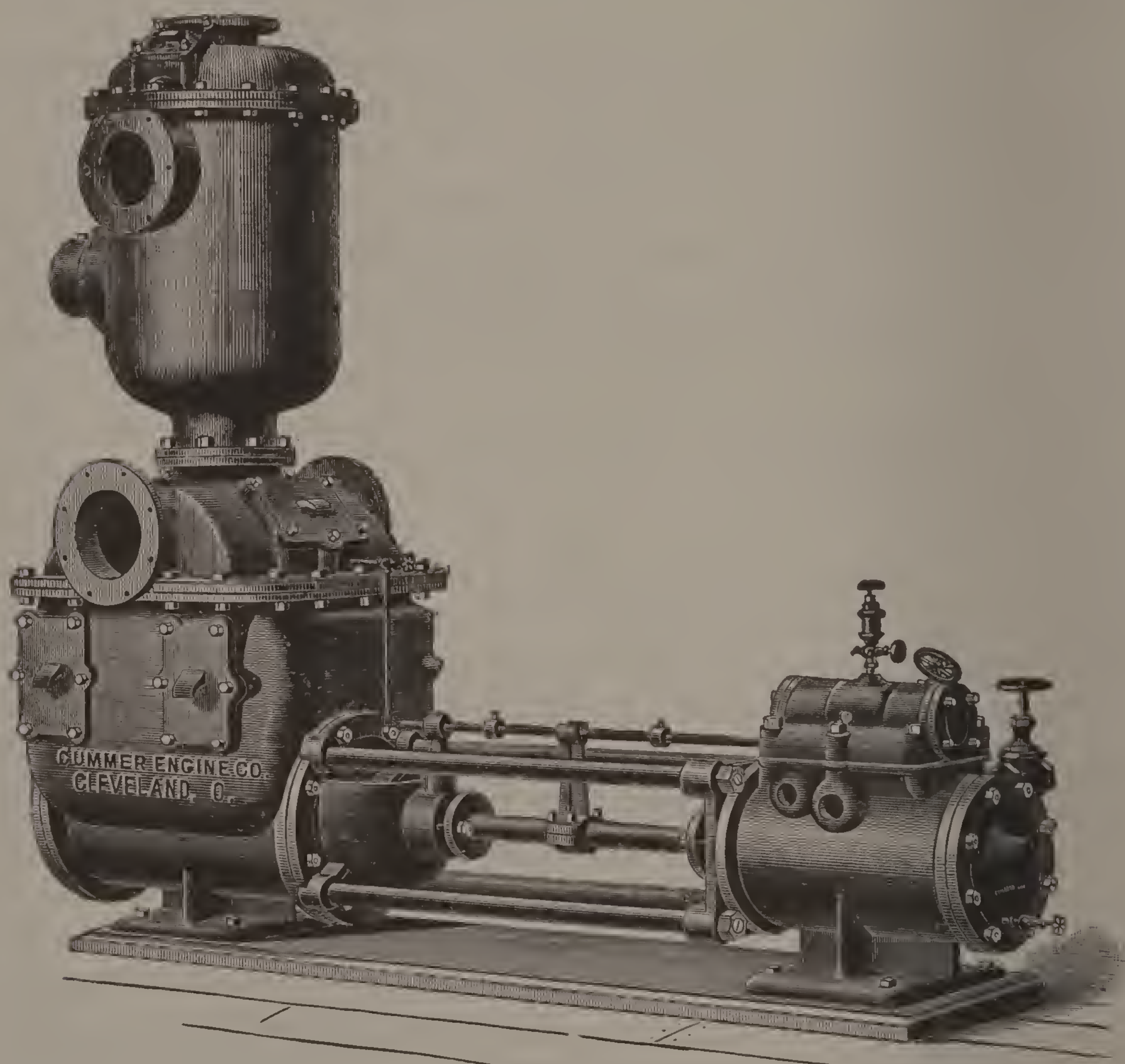


Fig. 27.

Whenever the height from the surface of water in the well, or other body of water from which the injection is taken, to centre of injection does not exceed about 20 feet, there is no separate pump for injection required, for as a vacuum is created in the condenser the atmospheric pressure forces the water into the condenser, where it enters in the form of fine spray.

Referring to Fig 27, the condensing apparatus is shown ready for attachment. At the right is the steam cylinder and to the left is seen the air pump above which is placed the condenser which is of cylindrical form. Two large side openings provided with flanges will be noticed, one in the air pump chamber, the other in the condenser. The upper one is to connect the condenser with the main engine exhaust pipe, while the lower one is for attaching the pipe which discharges the condensed steam and injection water and any air or vapor that may be in the condenser. There is a corresponding opening on the opposite side of the air pump, either one of these may be used for discharge as may be most convenient and the other one closed by means of a plate bolted to the flange. Injection water enters at the top of the condenser; there is shown a flange, above the side opening which admits the exhaust, and to this flange is attached the injection pipe. Exhaust steam from the steam cylinder on the right, enters the condenser through an inclined pipe running upward towards the left. There is a small pipe provided to inject enough water to condense this exhaust steam when the apparatus is starting. Having now described the essential parts, their action is as follows:—The air pump creates a partial vacuum in the condenser and exhaust pipe and passage, so that exhaust steam flows in with but very little resistance from back pressure. It is then met by a spray of finely divided water and is instantly condensed in which condition it is removed together with the water used for injection, by the further action of the pump and discharged to a hot well. Feed water may be drawn from this well if desired and the remaining water allowed to run away, if no further use can be made of it.

There is provided a safety attachment which prevents any water from reaching the cylinder in case the condenser should become flooded. The device works automatically in such a case and allows the exhaust to go to the atmosphere, while at the same time the vacuum is destroyed, it thus effectually prevents accidents such as sometimes occur when a condensing apparatus has no such provision against them.

We make a slight change in the above arrangement of condenser, in cases where we wish to get a high temperature of feed water, when using a condensing engine and employing two heaters in accordance with what has been already said in the article on feed water heaters. In this case, exhaust steam from the engine is led into a heater and thence passes through to the condenser, while the pipe which conveys exhaust steam from the pump is not allowed to communicate with the condenser at all but is connected with the second heater. Thus we take advantage of the high temperature of exhaust steam coming from the steam pump, which is now non-condensing, and make it yield further heat to feed water which has passed through the first heater but still has a lower temperature than

exhaust steam from the pump because exhaust steam from the main engine is much below atmospheric pressure.. These changes cause but a very slight modification of the above arrangement for condensing apparatus, while the resulting economy is great, and we strongly recommend this arrangement. An independent condensing apparatus such as above described is what we prefer to use with all important engines, but in other cases where a cheaper form of condenser is wished, we manufacture and furnish a condenser which may be attached directly to the main engine, its price while considerably less than the other independent arrangement is not enough less to recommend its employment with engines of any considerable size or importance, although for small and inexpensive engines, such a condenser adds much to the efficiency at very small cost. We shall not have space in this issue to illustrate by cuts this form of condenser, but its simple construction may be easily understood. The condenser consists of a cylindrical cast iron vessel, within which and at one side is cast the air pump, having fitted to it a bucket plunger with a trunk passing up through a large stuffing box and operated by a connecting rod attached close to the plunger. A condenser of this form is self contained and requires but very little space, it may be placed in any convenient position and need never be set below the floor level. The plunger having a trunk and short connecting rod, we may place the condenser under the main shaft and drive by means of an eccentric or we can drive from a disc crank on the end of the main shaft, by a belt from a pulley or any other arrangement that particular requirements may suggest. Where engines work at high speed which is unfavorable to the action of a condenser as ordinarily constructed, we have devised a special form of air pump, which may be given as rapid a reciprocation as the engine itself and still work smoothly and quietly to the entire satisfaction of ourselves and to those using our engines.

To facilitate the selection of a condenser, there is given, among the tables under each class of engines using independent condensing apparatus, a table which shows the proper condenser to use with each engine; the sizes are designated by letters as A, B, C, &c., from which letter, all the necessary dimensions may be found, by reference to the table of Independent Condensing Apparatus.

THE CUMMER ENGINE CO., CLEVELAND, O.

Independent Condensing Apparatus.

Letter Denoting Size.	AIR PUMP.			Diameter of Exhaust Opening in Cylinder.	Diameter of Injection Pipe.	STEAM CYLINDER.			DIMENSIONS OVER ALL.		
	Diameter Inches.	Stroke Inches.	Diam. of Discharge Pipe.			Diameter Inches.	Steam Pipe.	Exhaust Pipe.	Length.	Width.	Height.
A	8	10	4	This dimension will be made to suit the particular Engine to which it is to be applied.	2½ to 4	6	¾	1	6′ 9″	1′ 10″	5′ 8″
B	10	12	6		3 " 4	8	1	1½	6′ 11″	2′ 2″	6′ 0″
C	12	12	6		3½ " 4	10	1½	2	6′ 11″	2′ 2″	6′ 4″
D	14	18	8		4 " 6	10	1½	2	8′ 1″	2′ 8″	6′ 11″
E	16	18	8		5 " 6	12	2	2½	8′ 6″	2′ 8″	7′ 10″
F	18	24	10		6 " 8	14	2	2½	10′ 2″	3′ 3″	8′ 4″
G	20	24	10		6 " 8	16	2	2½	10′ 2″	3′ 3″	8′ 9″
H	22	24	12		7 " 8	16	2	2½	10′ 10″	4′ 9″	10′ 1″
I	24	24	12		7 " 8	18	3	3½	10′ 10″	4′ 9″	10′ 7″
J	28	24	14		7 " 8	20	3	3½	10′ 10″	4′ 9″	11′ 0″
K	30	36				20					
L	36	36				24					

In round numbers it requires twenty-five times as much water to condense the exhaust steam from an Engine as there was evaporated in the boiler to make it: that is to say, if an Engine is using 30 pounds of water per horse power per hour, there will be required for the condenser 750 pounds of injection water per horse power per hour.

It will be observed that two diameters are given for injection pipes. The first or smaller size is the one to be used in ordinary cases, but if the pipe is very long, or has many or abrupt turns in it, the larger diameter should be used to get the best results.

AN AUTOMATIC ENGINE ABOUT EQUAL IN PRICE TO A PLAIN SLIDE VALVE ENGINE OF EQUIVALENT POWER, EACH ENGINE FURNISHED WITH ITS OUTFIT COMPLETE.

It is sometimes questioned by manufacturers whether there will be enough saving in fuel to warrant replacing an engine of ordinary construction with a high grade automatic engine. We will endeavor to make this matter clear and to show not only considerable economy in coal consumption by using an automatic engine, but also for the same power a first cost for the whole outfit which is but very little if any more, and in some cases even less, than the cost of an outfit for an ordinary engine. This seems improbable at first sight, but it is nevertheless true; and, it results mainly from the fact that with an automatic engine we can use a higher pressure of steam and get it into the cylinder at nearly full boiler pressure so that for a given power, with an automatic engine, a smaller engine and cylinder may be used, and, in consequence of increased economy in the use of steam, we can use much less boiler power and still have all we want for the engine.

With a plain slide valve engine we cannot use a high boiler pressure and of this pressure only about $\frac{2}{3}$ is available in the cylinder in consequence of throttling and the improper proportions many makers give their valves and ports. Frequently the boiler pressure is reduced more than 50 per cent before it reaches the cylinder, as is daily shown by indicator cards from throttling engines. Such engines having no special means of cut-off have only a limited rate of expansion and hence the low economy resulting from these combined causes, as well as the lower horse-power which they develop. Now for the sake of comparison we will take one of our automatic engines Class B, and compare it with a plain slide valve engine, not of our own make, but selected from the catalogue of a prominent maker of good reputation. The automatic engine of ours which we will select, as it is a medium size, is a 14x24, the revolutions per minute are 140, steam pressure 90 lbs. developing 101 horse-power when cut-off at $\frac{1}{5}$ stroke and 121.2 horse-power at $\frac{1}{4}$ cut-off. The slide valve engine is also 14x24, revolutions 115, steam pressure 75 lbs. of which $\frac{2}{3}$ or 50 lbs. is available, point of cut-off $\frac{5}{8}$ stroke, horse-power developed 92.8.

From these data we calculated the pounds of water required to be evaporated for each engine to furnish the steam for the respective horse-powers per hour.

The quantity of water is as follows:—

Automatic	$\frac{1}{5}$ cut-off	requires	2361.57	pounds	of	water	per	hour.
"	$\frac{1}{4}$ "	"	2801.95	"	"	"	"	"
Plain slide valve	$\frac{5}{8}$ "	"	3130.108	"	"	"	"	"

These figures divided by the horse-powers corresponding to $\frac{1}{5}$, $\frac{1}{4}$ and $\frac{5}{8}$ cut-off give the pounds of water required for one horse-power per hour, or for the different cut-offs,

$\frac{1}{5}$ cut-off	requires	23.381	lbs.	water	per	horse-power	per	hour.
$\frac{1}{4}$	"	"	23.118	"	"	"	"	"	"	"
$\frac{5}{8}$	"	"	33.729	"	"	"	"	"	"	"

Since one pound of coal will evaporate 8 lbs. of water, the coal required per horse-power per hour is for,

$\frac{1}{5}$ cut-off with automatic engine,	2.9226	lbs.
$\frac{1}{4}$ " " " "	2.8897	"
$\frac{5}{8}$ " " plain slide valve,	4.2161	"

These figures correspond to a saving in coal by using an automatic engine of 30.7 per cent., with a cut off at $\frac{1}{5}$ and of 31.5 per cent., with a cut-off at $\frac{1}{4}$ stroke, which is a decided economy. The cut-off at $\frac{1}{4}$ shows the higher saving, because there are considerations which come in to modify the figure for theoretical economy for a cut-off at $\frac{1}{5}$, and it will be better to take a cut-off at $\frac{1}{4}$ as the more economical point, with a saving of 31.5 per cent. The coal required per day of ten hours for such an engine as the plain slide valve above is 1.9562 tons, and for one year of 300 working days 586.86 tons; an automatic engine would save nearly one-third of this amount.

In reality engines of each class consume more coal per horse-power per hour than the above figures show, because our calculation leaves out of account internal condensation and various small sources of loss which in the aggregate will modify our figures. A consumption of three pounds of coal of good quality per horse power per hour is a very good economical result, and with coal of poor quality the number of pounds becomes proportionately increased. But inasmuch as we have made the same assumptions for each engine the comparative result is the same; if it requires $3\frac{1}{2}$ pounds of coal for an automatic engine, then an ordinary engine will consume $4\frac{1}{2}$ or 5 pounds, the percentage remaining unchanged.

We have in several cases replaced ordinary slide valve engines by our own automatic engines, and effected a saving of 50 per cent., so that the above figures are much less than results obtained in actual practice. There has been an undue advantage given to plain slide valve engines in the above calculation; they will generally be found to cut-off later than $\frac{5}{8}$ stroke, and to suffer much more loss of pressure by throttling than we have assumed to be the case. In general, there is a saving of from 30 to 50 per cent., and we may safely say that there is a difference of 40 per cent. between the amounts of coal consumed by these two classes of engines. The saving in coal, however, is not the only consideration.

When we reduce the water required to be evaporated per horse-power per hour from 33.7 pounds to 23.1 pounds, it means that much less boiler capacity is required, and we save on the first cost of boilers. While we allow 15 square feet of heating surface to a horse power in ordinary engines, we only allow 10 square feet for an automatic engine. For a case where an automatic engine is used to displace a plain slide valve engine, and the same boilers are retained, there is an incidental advantage resulting from the less evaporation of water required. If a boiler, by reason of the increased economy of the engine, is called upon to evaporate 30 to 50 per cent. less water to supply a given power, it will follow that 30 to 50 per cent. less scale is deposited in the boiler. The life and safety of the boiler is thus increased, and the necessary care decreased; but, while this is a very important matter, we will not further consider it in the comparison which we are here making. In our calculation of economy, the 14x24 plain slide valve engine was found to yield 92 horse-power; this rating gives every advantage in point of economy to the plain slide valve engine. But such engines are not proportioned throughout with sufficient strength to develop so great a power, and they are rated by most makers at about 60 horse-power. In order, then, to make a fair comparison between the cost of an engine and boiler of each class, to furnish a given power, we will take the engine at its rated power of 60 horse, and provided with such a boiler, taken from the list, as the maker considers proper. We will then select an automatic engine of as nearly as possible equivalent power; this will be an 11x20 Class B engine, which, at $\frac{1}{4}$ cut-off, with 90 lbs. steam, gives 66 horse-power; and we will take from the same manufacturer's list a suitable boiler for this engine. The prices of plain slide valve engines, and of boilers, vary with different makers; but we have taken an average price, so as to make a fair comparison of the cost of a complete outfit for each class of engine.

Cost of 11x20 automatic engine	$1550.00
" 1 Tubular Boiler 48″ x 14 feet front, &c.	825.00
Cost of whole outfit	$2,375.00
Cost of 14 x 24 plain side valve engine	$1,200.00
" 1 Tubular Boiler, 60″ diameter, and 12 feet long, front, &c.	1,030.00
Cost of whole outfit	$2,230.00

RESULTS OF CALCULATIONS.

To summarize the results of the foregoing calculations we may say that for medium or large powers, with engines of the two types just considered, an automatic engine with its outfit may be furnished for the same or but slightly increased cost as that for a plain slide valve engine of equivalent power with its boilers. The cost may sometimes be even less, while the saving in coal amounts to from 30 to 50 per cent.

PISTON SPEED.

The piston speed of an engine is the distance in feet which the piston travels in one minute, and is, therefore, the product of the number of revolutions per minute by twice the stroke in feet. With small engines which have a short stroke, a high piston speed is had by adopting a high rate of revolution, while with the larger, long stroke engines the number of revolutions need not be nearly so great to secure the same speed. Thus, with our engines, a 6x12 engine has 200 revolutions and a piston speed of 400 feet per minute, while a 24x48 has 81 revolutions and a piston speed of 650 feet per minute. Piston speed, thus depending upon the length of stroke and rate of revolution, has to be determined with reference to each of these factors; for a certain piston speed, we may vary either the number of revolutions per minute or the length of stroke. But it is not good practice to go to an extreme variation in either direction; on the one hand, we must avoid an excessive rate of revolution. because then the influence of such reciprocating parts as the piston, connecting-rod, &c., has an injurious effect, besides the wear and tear and danger of breakdown, which is always a source of anxiety with engines having a high rate of revolution; while on the other hand, with large engines, where a long stroke and a slow revolution suffices to give a high speed, the long stroke makes necessary a very long frame and the low rate of revolution also, does not allow so well for proper regulation of speed. So it will be apparent that the size of an engine must be considered in determining what speed to give the piston. A high piston speed together with high rotative speed is desirable for several reasons; we can obtain in this way great power from moderate sized engines, and, since we must use considerable expansion for proper economy, a high speed is favorable to this in two ways: In the first place, it corrects, to a great extent, the evil of internal condensation, which is such a serious loss where much expansion is used, since it allows only a short time for the metal to part with its heat; and in the second place it permits a better provision to be made to meet the extreme variations in pressure which occur with expansive working; some means must be adopted to absorb the excessive pressure at the beginning of a stroke, and yield it up again when, later on, the pressure falls, and a high speed, which gives great momentum to the fly wheel without requiring excessive weight, is better adapted to do this than a slow speed. But it must be borne in mind that while, theoretically, a high piston speed and high rate of revolution is advantageous, there are practical considerations which limit their employment to moderate rates. We do not use, with our engines, an extremely high speed, but such moderate piston speeds as from 400 feet to about 650 feet per minute, and a rate of revolution of from 81 in our large sizes, to 200 in our very small sizes.

The speeds given in our tables we believe to be the best and most economical as regards first cost, subsequent repairs and consumption of fuel, and they may be depended upon for safe, satisfactory working. But we are not, by any means, obliged to confine ourselves for successful working to the speeds mentioned; our construction of engine and governor, our positive valve motion, and especially the fact that we make the reciprocating parts as light as consistent with proper strength, gives altogether such a combination that there is no limit whatever imposed upon the speed within a very wide range. Hence we are not under the necessity of advocating either high speed or low speed, in order to set forth the merits of our engines, because they will work well at any speed. We can therefore suit our customers and the requirements of each case. Our engine will run as fast as any high-speed engine and give satisfaction, or it may be run as slowly as the slowest rate of any other make and still work successfully. If a great deal of power is wanted from a small engine, we can supply one of our smaller sizes and run it at a high speed, or, if the conditions are such that high speed is not permitted, we can give the required power with a larger engine run at a slower rate.

INFLUENCE OF RECIPROCATING PARTS AT HIGH SPEEDS.

We may meet the variations of pressure which occur from expansive working in either one of two ways, by making the reciprocating parts very heavy so as not to need a heavy fly wheel, or to make the reciprocating parts as light as proper strength permits and depend solely upon the inertia of a heavy fly wheel to absorb and dispense the excess of power. In order to understand these two modes it may be well to briefly examine the principles upon which their action is based. By reciprocating parts is meant the piston, piston-rod, cross-head, connecting-rod and attachments, and, where these parts are designed to furnish a store of energy they are made unusually heavy, are given a high velocity and by opposing a resistance to the high initial steam pressure at the beginning of a stroke and giving out a pressure at the latter part of a stroke, the tendency is to equalize the variations of pressure caused by expansion with high pressure steam and an early cut-off. This effect is easily understood, a piston it is well known has a variable velocity; starting from rest at the commencement of a stroke it gradually increases to its maximum speed at mid stroke and then gradually decreases in velocity until brought again to rest at the end of a stroke. But in order to move a heavy mass from a state of rest and give it the variable accelerated motion and high velocity at mid stroke which a piston has, it is necessary to continually exert a variable force; and to bring the mass to rest again there is given out a resistance to being retarded, which is also variable, and corresponds to the force which first set the weight in motion. Now, the force

required to give our reciprocating parts their accelerated motion, is taken from the steam pressure or from the stored energy in the fly wheel and is, therefore, just so much subtracted from the effort to move the crank at the commencement of a stroke, while the resistance opposed by the reciprocating parts to being brought to rest, is expended in pressure to move the crank during the latter half of a stroke.

For a certain steam pressure, cut-off and rate of revolution, it is possible to so fix upon the weights of moving parts that the pressures on the crank pin become nearly equalized, but these weights are correct only for the conditions taken; if the steam pressure, cut-off or speed varies, the engine will not work smoothly. Moreover, it is a very nice calculation to correctly apportion the required weights, and when we depend upon this means of regulation, we are obliged to use a combination of great weight and high speed, this introduces a disturbing influence which designers generally seek to avoid. Whenever there is rapid reciprocation in machinery, it means considerable jar and shock, which may be diminished by keeping within reasonable speed and having the moving weights as light as possible, but here we find the difficulty purposely aggravated and the effect cannot but be injurious; especially is this the case with small engines which necessarily have a high rate of revolution, and, it is well known that small engines constructed upon this plan are unsuccessful. This system of regulation requires that we move a heavy weight from a state of rest, give it a high velocity and then bring it to rest again; and, when this has to be done 300 to 600 times a minute, it is evident that a serious strain is brought upon an engine and that the risk of breakdown becomes very much increased. It may be seen also, that the resistance which the piston and connections opposes to being set in motion at the beginning of a stroke, can be great enough to absorb the whole steam pressure exerted at this point, and although this pressure is given back later on, yet such a state of things sometimes causes very irregular action. Weight in the reciprocating parts is really an evil which is only increased by making them unnecessarily heavy and giving them a high speed. A familiar example of this is seen in the locomotive which, for passenger traffic is a high speed engine, the revolutions per minute are 350 and upwards, but so great is the strain exerted upon the crank pin, connecting-rods and side-rods, from such rapid motion, that accidents have frequently occured, and the effort is made to keep the reciprocating parts light and as strong as possible. The same thing should be done with stationary steam engines, for, although a regulation can be effected by rapidly moving and heavy reciprocating parts, yet this regulation is only partial and imperfect, besides having decided disadvantages; we can accomplish all that is desired by means of a fly wheel alone and introduce with it no disturbing element to regular action.

THEORY OF FLY WHEELS.

When a fly wheel is used, instead of having heavy weights moving back and forth with a variable velocity, there is a wheel with a heavy rim revolving continuously in the same direction with great velocity and thus storing up a large amount of energy. The work done by the steam at the commencement of a stroke, in excess of the mean work, is so small compared with the energy stored in the fly wheel, that no great increase in velocity can take place: the fly wheel simply absorbs the excess without a perceptible increase of speed. A similar state of things exists at the latter part of a stroke, when the steam pressure has rapidly fallen and less work is done; the fly wheel then yields up a part of its energy to supply the deficiency and still does not visibly retard its speed. The varying effort of the crank pin and the varying resistance of the load is also met in the same way, and the result is to approximate very closely to a uniform tangential pressure upon the crank pin, with only very slight variations in the velocity of the fly wheel. Just at this point comes in the office of the governor, for the slightest change of speed is instantly met by a change in the point of cut-off and standard speed restored. But it is important to have a fly wheel so proportioned that it does not admit of any sudden or great change of speed. If the rim is not heavy enough, any slight change of pressure or load causes considerable change of speed, and when the governor acts to correct this, there is caused a variation in the other direction; thus with a sensitive governor and a light fly wheel, there is a continual and quite unnecessary fluctuation of speed. This does not occur with such weight of rim as we have adopted with the fly wheels for our engines; with them the permissible variation from standard speed is very small, and we depend upon our governor for the rest of the regulation; the result is an engine which varies less from a fixed number of revolutions per minute than any other now in the market.

FLY WHEEL DIAGRAMS.

As a matter of interest in connection with fly wheels, we have made some diagrams which show the variations of power, above and below the mean work done in a revolution, with a steam pressure of 90 lbs. and a cut-off of $\frac{1}{5}$ stroke in one case, and a cut-off of $\frac{3}{8}$ stroke in the other. The mode of constructing these diagrams is very simple and may be briefly explained:

We constructed an ideal diagram, such as Fig. 28, for steam at 90 lbs., cut-off at $\frac{1}{5}$ stroke, and then expanded to the end. The length of the diagram representing the stroke, we may construct a semi-circle upon this line and divide the half circumference into equal parts, representing successive positions of the crank; if the connecting-rod be infinitely long,

lines drawn from these points, at right angles to the base line, will give the corresponding positions of the piston, and, being ordinates of an indicator diagram, will show by their length the pressure upon the piston for each of these positions of the crank. These pressures, however, are not just those which propel the crank; it is the tangential pressures upon

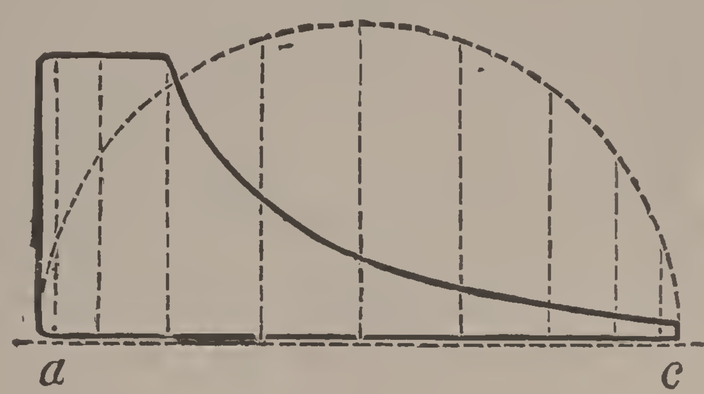


Fig. 28.

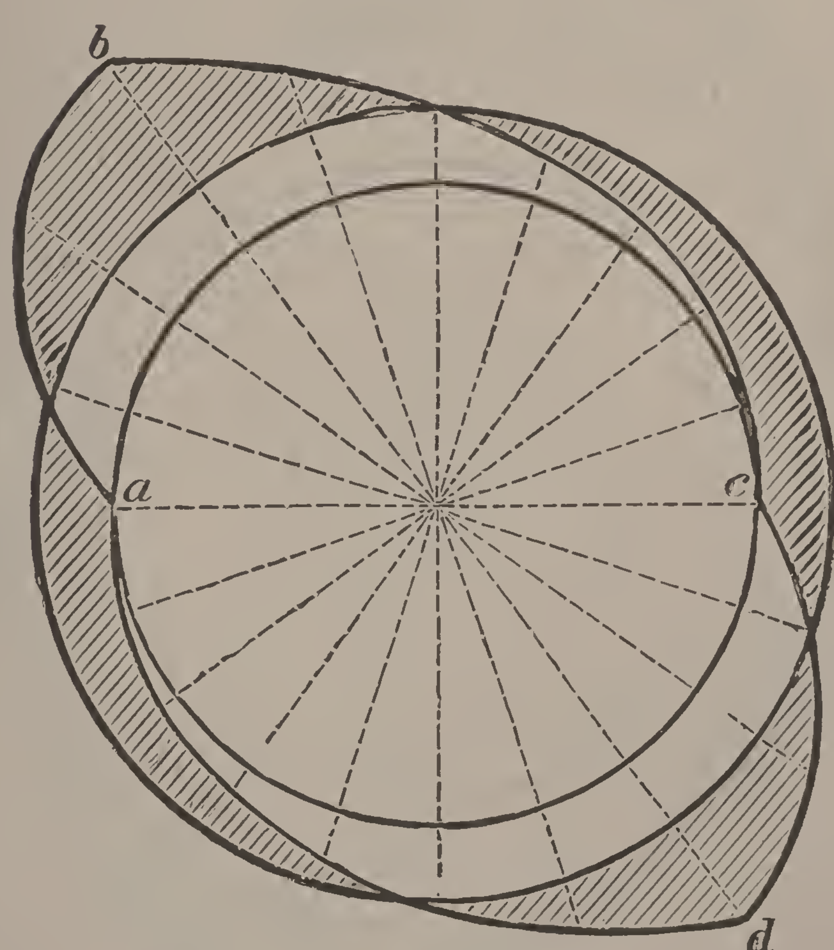


Fig. 29.

the crank which we desire to have; these are different for each angle of the crank with the line of centres; and, since in practice the connecting-rod has a definite length, its modifying influence must be allowed for. We made then a table of tangential pressures for forty equal divisions

of the circle, supposing the radius to represent a pressure of one pound per square inch on the piston, and allowed for a connecting-rod of six cranks' length. Then from the diagram was found the effective steam

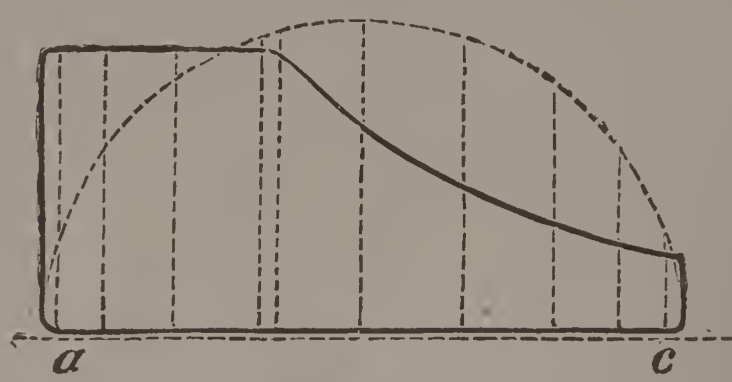


Fig. 30.

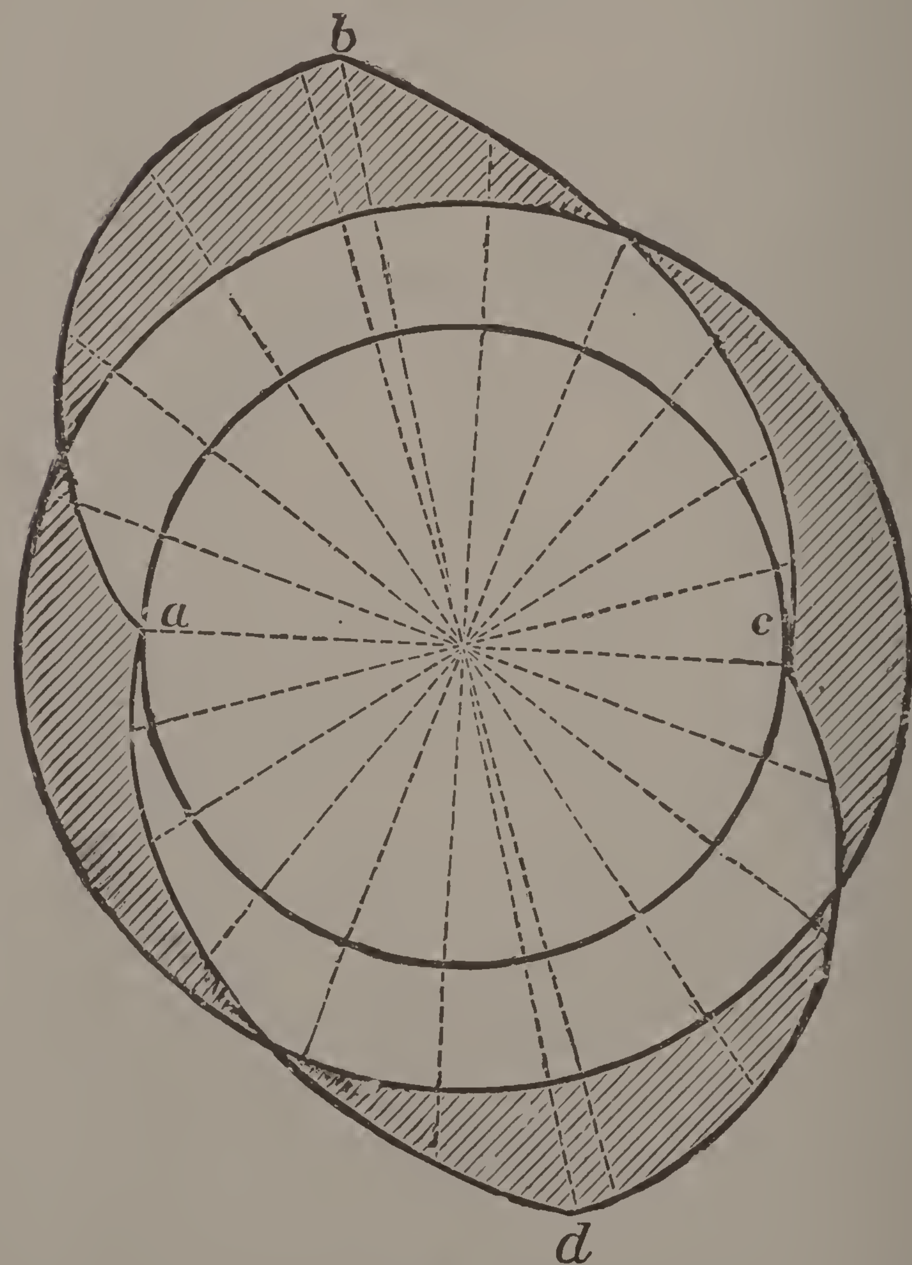


Fig. 31.

pressures corresponding to each crank position, which, multiplied by the tangential crank pressures for one pound pressure on the piston, gives

the tangential pressure for each division, with 90 lbs. initial steam pressure, and, the various effective pressures for successive points as taken from the diagram. Referring to Fig. 29, the inner circle has a diameter equal to the length of the diagram, and it is divided into a number of equal parts, and radial lines are drawn extending beyond the circle; upon each line is laid off the the corresponding tangential pressure, and the extremities joined by a line makes the curve a b c, starting from a, where the pressure is 0, and ending at c, where the pressure is also 0. The return stroke gives a corresponding curve, c d a. We now find the mean tangential pressure, which is, of course, the average of all these other pressures, and represent it by the outer circle. It will be seen that the area enclosed between the two circles represents the work done in a revolution; also that the sum of the areas included between the two curves, a b c, c d a and the inner circle, represents the same work. These two curves cross the circle of mean pressure, a part of the curve being above, and a part below the line; for that part of a revolution where the curve extends above the circle, the work is in excess, and for that portion where the curve goes below, there is a deficiency of work done. Two phases of a revolution show an excess, and two phases a deficiency, and the excess balances the deficiency for a whole revolution, as we find on calculating the areas. Comparing the whole excess or deficiency with the work done in a revolution, which is represented by the mean pressure on the crank pin, exerted through 360°, or the complete circle, we find a variation of 37.5 per cent. for a whole revolution. If we take the greatest area, which is the excess during the first quadrant, and compare it with the work for a stroke, or a half revolution, we have a variation of 45.4 per cent. Similar diagrams, such as Fig. 30 and Fig. 31, were constructed for 90 lbs. steam pressure and a cut-off at $\frac{3}{8}$ stroke; these give a mean variation of work above the average of 29.2 per cent., and a maximum variation of 32.8 per cent.

If our practice was to use heavy reciprocating parts and excessively high speeds, these figures would have to be modified in accordance with what has been previously stated about the influence of these parts at high speeds; but since these parts are made light, and moderate speed is adopted, with a cut-off at about $\frac{1}{4}$, we may take a variation of 40 per cent. as about right for the conditions; and, the formula we have constructed for fly wheels provides that, with a variation of 40 per cent. above the mean work done in a revolution, there shall be only $\frac{1}{40}$ of a revolution variation in the speed. We do not mean from this that our engines show anything like this variation in speed when working, but only that such a limit is established for the fly wheel, and, that these fluctuations, together with such as occur in the load, are controlled by the governor so as to secure an unusually steady speed for the engine.

IMPORTANCE OF A STEADY POWER FOR FLOURING MILLS.

While with electric lighting thc importance of a steady speed is so well recognized that the remarkable uniformity of the Cummer Engine at once favorably recommended it and has secured its adoption by several of the Electric Light Companies, and while with cotton and woolen mills a high grade automatic engine is invariably employed because of the imperative necessity for uniform motion, it is no less important that the engine for a flouring mill should have a very regular speed, although the necessity may not be so clearly seen as in the other cases mentioned. Milling is now a great industry, and with growing competition and better methods, any means of securing a better product and at less cost is worthy of attention. For economy in power a high grade engine must be used, and for uniformity of product and economy of manufacture (good and uniform grades and good yields,) a regular speed is of the greatest importance. The separation of the product from the rolls into different grades and from impurities, is now largely effected by means of a blast, and, the perfection of the operation of this blast is much more dependent upon the speed of the engine than is ordinarily imagined. The blast and slides once adjusted and everything working properly, the process will go on regularly so long as the same speed is maintained, but just as soon as the speed changes, the grade of the flour and the richness of the middlings, bran and other impurities will be affected ; we get either less flour, or flour of poorer quality, unless the slides are re-adjusted. But it is impracticable to change the slides for every variation of speed, and these variations might not always be noticed though none the less producing an injurious effect. These facts though, and their importance are generally understood by the intelligent miller, and he will also assent to the statement that the engine which while saving in fuel comes also the closest to absolute uniformity of speed, is the one which will make the most flour of the most uniform grades and at the least cost.

For large mills, making from 500 to 1000 barrels of flour a day, an engine of unusually good economy will effect a very great saving, and, the consideration of whether 20 or 30 pounds of coal is to be used to produce a barrel of flour, becomes an element in the cost of production which it does not pay to disregard. An engine of the highest grade costs more money, but the great economy of fuel so secured makes it profitable to employ one in a large mill, where considerable power is required, rather than to use a lower priced but more wasteful engine. For large mills we recommend a compound condensing engine, steam jacketed, and provided with every means to prevent loss of heat ; we will undertake to furnish an engine of this kind which will produce a barrel of flour for each 20 pounds of coal consumed, and, even better results than this may be expected from such an engine.

STEAM JACKETS.

There is a decided economy derived from using a steam jacket, although the advantage has no doubt been frequently overrated. The prejudice in some minds against them results sometimes from an imperfect understanding of the subject, and, sometimes from an unfortunate experience with a steam jacket which has not been properly applied. When a steam jacket is correctly used, the space between it and the cylinder is kept filled with steam at boiler pressure; exhaust steam will not do because it has too low a temperature; nor may there be any communication whatever with the inner steam cylinder without losing the advantages sought. It is highly important also to make proper provision to draw off the water formed by steam which condenses in the jacket; and this water should for economy be led into the heater, and its heat returned to the boiler. A steam jacket may be rendered well nigh useless if water is allowed to accumulate and remain in contact with the cylinder, the space must be kept filled with live steam.

The reason some engine builders have failed to secure economy with a steam jacket is, that points such as those just noted have been either not fully understood or else they were neglected. In some cases which have come under our notice, there has been an intentional communication between the steam jacket and the engine cylinder; in many other cases, failure has resulted in consequence of having cracked cylinders, which caused unwittingly the same bad effect. We are thoroughly familiar with the proper and necessary construction for steam jacketing, and have invariably met with full mechanical and economical success, whenever we have used this invaluable adjunct in our practice.

With compound engines a steam jacket should invariably be used for both high pressure and low pressure cylinders. Considerable economy also results when a steam jacket is applied to a single cylinder engine whether condensing or non-condensing, the advantage however being less marked with the latter than with condensing engines. The size and importance of an engine will of course largely determine whether a jacket may profitably be attached.

A steam jacket is designed to prevent those losses from initial condensation and condensation during expansion which occur in an unjacketed cylinder when much expansion is used, thus giving rise to extreme variations in temperature. These losses become greater as the grade of expansion increases, and there is a point beyond which expansion even with a jacketed cylinder ceases to be economical. To understand how a steam jacket acts to prevent initial condensation and condensation during expansion, we will consider what takes place with steam in an ordinary cylinder. When steam is first admitted to the cylinder,

it meets with surfaces which have been cooled by communication with the condenser; no work can be done by the entering steam until the metal of the cylinder attains the same temperature as the steam, and, condensation must take place until an equality in temperature is established. After cut-off there is a further condensation, because, whenever dry, saturated steam expands doing work, a portion of it becomes liquified. Thus, in an unjacketed cylinder, these two causes operate to bring about the presence of a quantity of water, which is deposited as a film of moisture on the interior surfaces, and is also dispersed throughout the whole body of steam in the cylinder. Part of the sensible heat of this water is given up to the steam while expansion is going on, and part of the water itself is re-evaporated as the pressure falls, and thus allows steam to be generated at a lower temperature and pressure; but, the larger part is still present as moisture when the exhaust valve is opened. At this moment then the cylinder is filled with wet steam and a quantity of water which has not been able to evaporate; but, just as soon as the valve opens the pressure is relieved, and the water then passes into steam of a pressure corresponding to that in the condenser abstracting from the cylinder as it does so, a large part of the latent heat necessary for evaporation; the wet steam also, which is an excellent conductor of heat, expands into the condenser, and takes away a large quantity of heat from the cylinder. Now this clearly is all lost heat, because no useful effect whatever is produced by the steam so condensed, and the cylinder is cooled much more during exhaust than would be the case if no water were present but only dry steam. It is necessary also to again heat up the cylinder before any motion can take place for a return stroke and the heat so needed must come from the entering steam, causing a portion of it to be condensed.

For the best economical working, then, it is plainly necessary to prevent, as far as possible, any condensation of steam either at the period of admission or during expansion. High speed, which allows but a very short time for any transfer of heat to take place is a very excellent way to lessen loss from this cause; but, principally may we prevent loss from condensation by using a steam jacket. When a steam jacket is employed the cylinder is kept always at the same temperature, which is at least as high as that due to the initial steam pressure. In this way there is no initial condensation, nor is there any condensation during expansion, since the quantity of heat which disappears doing work is supplied by the jacket, and the steam is kept saturated. At the time of exhaust opening when connection is made with the condenser, the steam expands as before, but it is now dry, saturated steam, which receives and parts with heat slowly, so that it does not abstract as much heat from the cylinder when expanding into the condenser, as did the wet steam in

the former case ; there is also no water or moisture in the cylinder to be re-evaporated as soon as pressure is relieved, and so although the steam jacket does supply enough heat to prevent liquifaction, and also heats up the cylinder from the temperature due to the exhaust to that of the entering steam ; yet, this quantity of heat is much less than that which is extracted when the cylinder is unjacketed.

It is not correct to say that steam in the jacket is condensed without doing any work, it does perform work because the heat units supplied correspond to that heat which disappears for the performance of work in the engine, and which causes liquifaction in an unjacketed cylinder; there is also supplied the quantity of heat required to make good that extracted during exhaust, and which otherwise would be just so much taken from the effective work of the steam in the engine. Recent experiments made with engines both jacketed and unjacketed, have shown a saving of from 6 to 18 per cent., according to the grade of expansion, in favor of jacketed cylinders. For ordinary cases we may count upon securing an economy of about 10 to 12 per cent., and this with large engines is of enough importance to warrant the use of a steam jacket.

COMPOUND ENGINES.

Opinions differ in regard to the utility of compound engines, but in spite of the objections sometimes urged against them, they have grown into favor especially of late years and now there are very few large marine engines or large engines for water works or in other situations where high economy is desirable, which are not constructed upon the compound system. In large cotton and woolen mills, flouring mills and all places where considerable power is required it will be found advisable to use a compound condensing engine, since this gives when properly designed the highest economical results ; we recommend these engines for such requirements and are prepared to construct them when desired by our customers. We have had a large experience with this class of engines and have secured marked economy in many cases where compounding has been used. We purpose publishing later same data derived from actual practice which will show very clearly the economy of compound engines, such as we have constructed.

The increased economy resulting from using a compound engine is mainly in consequence of the higher expansion which may thus be secured. With a single cylinder engine there is an early limit to economical expansion beyond which it does not pay to go. But to obtain the highest economical results it is necessary to get more work out of the steam than this limited range allows, and, it is just at the point where expansion in a single cylinder ceases to be profitable, that a second cylinder may be

employed to carry the expansion further. The serious loss by internal condensation and re-evaporation in a single cylinder engine using high expansion is in consequence of the great range of temperature between that due to steam of initial pressure and the temperature of steam exhausted into the atmosphere, or into the condenser. With a compound engine this range of temperature is divided between two or more cylinders, but generally two, one of which is a high pressure and the other is a larger low pressure cylinder. Steam in the high pressure cylinder may be worked to as high a grade of expansion as is found economical, and is then exhausted into the low pressure cylinder where it is further expanded to whatever amount is deemed advisable. In this way the small cylinder works between limits such as occasion but slight loss of heat from condensation and the large cylinder works between the temperatures of exhaust steam from the high pressure cylinder and the temperature of the condenser; these limits not being widely separated, there is not that great variation of temperature in the cylinder such as is found with a single cylinder engine, and thus, the loss by internal condensation and re-evaporation is very much less; hence the economy of compound engines; for, in this way the economy which directly results from high expansion may be secured without having those great losses which would occur with steam expanding to the same extent in only one cylinder. It is an excellent plan to use an intermediate steam jacketed receiver, into which the high pressure cylinder exhausts and from which the low pressure cylinder takes steam as if from a boiler. The second cylinder then has an expansion valve and works as a low pressure condensing engine. This method is to be recommended because in this way the great loss of pressure between the two cylinders as ordinarily placed may be avoided and we also have dry steam for the engine. Another great and unquestioned advantage obtained with compound engines is the better equalizing of the working strains which may be effected when two or more cylinders are employed. When much expansion is used in a single cylinder there is a wide difference between the steam pressure at the beginning of a stroke and that which the steam has at the end of a stroke. The strength of the engine and all working parts has always to be such as to resist the maximum pressure which is that of the steam before cut-off takes place, and, in a large engine this pressure may be very great and make necessary unusual strength and weight for the engine and all working parts. With a compound engine these strains will be very much lessened because the high pressure cylinder for an engine of the same power will be made much smaller, which greatly reduces the total initial pressure, and the low pressure cylinder can be arranged so that the strains are more nearly equalized throughout a stroke, than could possibly be done with a single cylinder engine. The variations in pressure being

thus practically lessened, a smaller fly wheel may be used, and the engines themselves need not be so heavily built. The extra strength required for all parts of a single cylinder engine, when expanding many times is one of the principal considerations which limit its employment to low rates of expansion, but with the compound engine we largely do away with this difficulty, as well as lessen the great loss of heat which is consequent upon much expansion in a single cylinder. In view of these advantages, and from the results obtained in our own practice, we consider that a compound condensing engine is the best form that can be adopted for engines of large size, where great economy is desired.

QUANTITY OF FEED WATER PER HORSE-POWER.

The theoretical quantity of water required per effective horse-power per hour depends upon the efficiency of the engine, varying with the steam pressure and rate of expansion and proportion of back pressure assumed in the calculation. The quantity of water so found is very much smaller than it is ever possible to realize in practice. The quantity as calculated from the terminal pressure of an indicator diagram is also less than the actual quantity consumed and is not to be trusted as anything more than a comparative figure. So that, we prefer to use in our calculations, for the amount of water requiring to be evaporated with good economical engines, a figure based upon the average conditions in practice; this is 30 pounds of water per indicated horse-power per hour. Engines of high economy will use less than this amount, and some of the wasteful forms will require considerably more but, for most cases in ordinary practice 30 pounds may be safely used, for a cut-off varying from $\frac{1}{4}$ to $\frac{1}{2}$ stroke; and, the margin allowed will cover the various small losses not provided for in the purely theoretical calculations. To find on this basis the quantity required by any engine of our several classes, it is only necessary to refer to the table for that class, and, to multiply the horse-power for the given steam pressure and point of cut-off by 30 pounds, the result is the weight in pounds to be evaporated for that horse-power. Thus, an 18x36 class C engine develops with 90 pounds steam at $\frac{1}{4}$ cut-off 214.7 horse-power and therefore 214.7x30=6441 pounds. But since the engine works beyond this cut-off, it would be better to calculate the water from the horse-power developed at $\frac{3}{8}$ cut-off. Thus 286.4x30 = 8592. pounds of water. Sometimes it is required to reduce the pounds of water to cubic feet or to gallons, and for such cases we give below the necessary data. For purposes of computation a cubic foot of water may be taken to weigh 62.5 pounds and a United States standard

gallon to weigh 8.34 pounds; a gallon contains 231 cubic inches, one cubic inch weighing 0.0361 pounds.

Taking the pounds of water as in the above example, 8592÷62.5= 137.4 cubic feet; and 8592÷8.34=1030 gallons of water for a cut-off at $\frac{3}{8}$ stroke.

QUANTITY OF COAL REQUIRED PER HORSE-POWER.

We have already referred to the quantity of coal consumed per horse-power per hour, but will repeat it in this connection, since the amounts of water and of coal ought to be considered together.

The economy of an engine depends greatly upon its size; small engines are more wasteful than large ones; all the items of loss become proportionately greater as the size is decreased, and, it is also more difficult to guard against waste with small engines than with large engines. With a good automatic, non-condensing engine, the coal required per horse-power per hour varies from 3 to $3\frac{1}{2}$ or 4 pounds, according to the quality of the coal. A condensing engine of the automatic type, will go somewhat below these figures. Still better economical results than these may be obtained with higher rates of expansion, using a compound condensing engine of moderately large size, employing a steam jacket and adopting every means to prevent waste of heat; the consumption of coal may then be made to go as low as $1\frac{3}{4}$ to 2 pounds per horse-power per hour; and, future improvements may be expected to bring about even higher economy than this, which, still falls far short of what is theoretically possible.

FORMS OF VALVES IN ORDINARY USE.

The kinds of valves in ordinary use for stationary steam engines may be classed, with sufficient exactness for our present purpose, under the heads of piston valves, rotary valves, and flat valves. There are many varieties of each kind of valve, particular makers adopting whatever valve is best suited to his use, and introducing such changes in the general form as his own special requirements may demand. It is important always to adopt that form of valve which gives the best results and which, while easily constructed, is the most durable in use, as well as economical in repairs whenever these are necessary. Beyond question, flat valves satisfy these requirements better than either piston valves or rotary valves, or any other kind of valve which has yet been devised, and they should be used to secure the best results. But it may happen that some feature in an engine will preclude the choice of a valve of this kind. Thus the governor may not have sufficient power to control an ordinary flat valve, and hence there must be used some form

of balanced flat valve, with its objectionable features, or else there must be employed a piston valve or a rotary valve, either of which is also balanced and requires but little power to move it, but is defective in point of durability and efficiency, as we shall show later. We have had a large experience with all these kinds of valves, and have come to the conclusion, that a plain flat valve—such as we will presently describe—is decidedly the best kind. Our governor being such that it is well able to control valves of this form, we were entirely free to choose whatever we considered to be best. We have, therefore, adopted for our engines plain flat valves ; and, in consequence of the peculiarities of our cylinder construction, and by making our valves very small, with several openings through them to admit steam, thus securing a large port opening with but a small movement, we are able to reduce the travel and the friction to a very small amount ; thus the power required to operate them is greatly reduced, and the governor is able to perfectly control the cut-off valve without the necessity for any special means of balancing. Since balancing is a serious complication, which is only effective for a short time, because such valves will not long remain tight, the advantage of our valve arrangement, which avoids these defects, and secures the best form of valve, is at once apparent. An automatic engine, belonging as it does to the highest grade of engines, ought to be fitted with the best form of valve ; other things being equal, that engine which is enabled by its construction to employ the best valve, is the best engine. We have elsewhere set forth, at length, the peculiarities of our valve and cylinder construction, such as is used with our automatic engines, and the advantages of such a system, together with the advantages of a positive connection of the governor with the cut-off valve, need not further be considered here. But the reasons for adopting this kind of valves and their superiority over the ordinary plain or balanced valves, piston valves and rotary valves, as well as the superiority of our automatic engines over the plain slide valve variety, will be more apparent when we examine them all in some detail, as will be done in the following articles on these valves.

THE PLAIN SLIDE VALVE.

In our article on the cylinder and valves, we have already alluded to tne fact that the duty of admitting steam to a cylinder, of cutting off steam, and of finally exhausting it, is an action entirely too complicated to expect from the performance of a single valve, and that the correct action for one requirement will not allow the proper action for the others. The ordinary plain slide valve may be made to cut off by giving angular advance, and adding lap to the valve ; it then becomes necessary to give lap to the exhaust side, where an equal amount is given the

steam is cut off, and the exhaust is closed at the same instant, in other words, compression and expansion begin at the same time; by allowing release to take place somewhat before the end of a stroke, the compression may be delayed; but, even then, compression occurs so early that a single valve cannot be advantageously made to act as a cut-off earlier than about $\frac{5}{8}$ stroke, so that a plain slide valve engine can only begin to cut off at a point which is later than it is ever considered economical to go with an automatic engine, or an engine with fixed cut-off. Our automatic engines have an economical range of from $\frac{1}{5}$ to $\frac{3}{8}$; but $\frac{3}{8}$ is the maximum point of cut-off ever advised by us; although they can follow up to $\frac{7}{8}$ stroke, and in this respect, are unlike any other automatic engine in having a range of power from 0 up to $\frac{7}{8}$, which may all be used if wanted. Most automatic engines have their limit of cut-off inside of $\frac{1}{2}$ stroke, and beyond this the majority of them cannot go; our engine, however, admits of a cut-off at $\frac{7}{8}$ stroke, and although this cannot, of course, be used with economy, yet, it may often be very useful and desirable in cases where there is occasionally an unusually heavy load upon the engine. Since plain slide valve engines have the limited range of cut-off of from $\frac{5}{8}$ to end of stroke, they can only be used with advantage for small powers, or in cases where economy of fuel is secondary to cheapness in first cost. These engines are simple in construction; they consist of but few parts, and the low price for which a small engine of this kind can be furnished, has caused them to be largely employed. With the larger sizes, as we have elsewhere pointed out, the cheapness in first cost does not commend them, because the cost of a complete outfit approaches so closely to that of an automatic engine with its outfit, that when economy in fuel is to be considered, the difference in favor of the latter engine is too great to admit of any question as to which is the better one to adopt. But plain slide valve engines, even wasteful as they are admitted to be, need not show so extravagant a coal consumption as will be found with most engines of this class in ordinary use. The card No. 7 shows a loss of power in consequence of a faulty valve and port construction and bad governor, such as is by no means uncommon. The card No. 8 is from the same engine after changes had been made by us to remedy these defects; in the first card, out of 104 pounds boiler pressure, the highest mean effective pressure that could be obtained in the engine was 56 pounds; but, in this latter card, we secured a mean effective pressure of 65.3 pounds from only 93 pounds boiler pressure, and increased the ultimate horse power developed from 273 horse-power to 318.5 horse power.

PISTON VALVES.

Piston valves, as their name would imply, are of cylindrical form and slide lengthwise back and forth within a cylindrical passage in which is

situated the ports. The opening edges of the valves and ports bear the same relation to each other as they would in flat valves, only, instead of being upon a plane surface they are upon a cylindrical surface. The action of the two valves, when properly seated and in good condition, is similar, but the questions of durability and continuous economy give rise to a very important difference, and constitutes a serious objection to a piston valve. With a flat valve, even a considerable amount of wear, provided it be equally distributed, will only serve to make a better fit between the valve and its seat, but a piston valve being cylindrical, and its seat also being cylindrical, it is evident that the slightest wear that would make the valve smaller and its seat larger must destroy the perfection of the fit and therefore impair the tightness of the valve, and the difficulty is only aggravated by increased wear. The injurious action of even the least wear with a piston valve arises from the nature of the surfaces in contact, and cannot well be overcome by any mechanical means; the whole length of the port must be at times covered by the valve, and the surfaces must be in the closest contact, but this cannot possibly be done if the valve or its seat has worn in the least from true cylindrical form. All valves must wear when in continued use even under the most favorable conditions. But in the great majority of cases where an engine is used, the conditions are decidedly unfavorable, and a good engine ought to be adapted to work well under all such circumstances as are ordinarily met with. One of the most trying agents to cause the destruction of valves is bad water. For some few favored localities, where the water is excellent, and for marine engines where surface condensers give pure distilled water for the boilers, an engine with a nicely fitted piston valve may work very well for quite a long time, or until a leak is caused by friction alone; but with engines for the country at large, we have to deal with all kinds of water, which may contain salts of lime, magnesia, and various impurities; while in many cases where water comes from driven wells and muddy streams, a very fine sand will be mingled with the water; these substances all find their way over into the cylinder, and get between the valves and their seats; evidences that such impurities are brought over by the steam and entrained water, may be seen in the deposit of a light gritty substance around bolt heads and in places where steam has leaked through, and it is shown by its presence in the cylinder and steam chest, but most unmistakable evidence is seen in the wear of the valves themselves under the grinding action of this grit. Such wearing action is bad enough in itself, but steam has even a worse cutting action. Flat valves, unless they wear unevenly, will stand a very large amount of wear before any steam can blow through, but with a piston valve, the moment it wears, that instant it begins to leak, and when steam gets a chance to

blow through even a small opening, the opening is rapidly enlarged by the friction and cutting action of the steam. A mere pin hole in a boiler, for instance, if not stopped up, becomes from this cause very much larger in a short time; the same action operates to destroy piston valves, or any form of valve which will not admit of wear without leaking. One of the most troublesome things to deal with where piston valves are used is the unequal wearing of both valve and valve seat, not only does the valve wear out of true, but its seat also departs from true cylindrical form; flat valves, when worn so as to leak, can easily be refitted by scraping, but with a piston valve any scraping or other operation to true up the valve only makes it that much smaller, and with the valve seat a similar operation renders it larger than before, consequently the fit is lost. Therefore, when repairs are needed, nothing else can be done but to make a new valve, and, since even this will not fit the old seat, the valve seat when much worn, has to be rebored, which is an expensive and troublesome operation. A mode of construction which secures durability in the parts of an engine, and therefore secures economy in repairs, is a matter which is scarcely less deserving of consideration than that of economy in steam. True economy does not mean merely cheapness in first cost of the engine and entire outfit, nor does it consist only in economy in coal consumption, it includes also the cost for attendance and repairs and all the running expenses, not merely for the first year, but for ten years, or a longer period, of continuous efficient service. The question of durability is fully as important a factor in economy as that of first cost or of coal consumption, and must be considered carefully before any correct judgement can be formed about the actual economy of an engine. A common slide valve engine may be much more truly economical than even an automatic engine, if the latter, through any defective principle, requires frequent and expensive repairs. The valve is the vital part of an engine, and anything which affects its durability or impairs its efficiency will affect the whole engine to a greater extent than the failure of any other part. Any form of valve then, such as in itself is liable to easily get out of order, or to fail to act properly under such conditions as are met with in the majority of cases in practice, should be studiously avoided, and the preference given to that form which, as in the case of a plain flat valve, works satisfactorily, when properly designed and proportioned, under all the ordinary conditions that may arise. The only advantage claimed in favor of piston valves is that they are balanced, but this advantage, as in the case of other balanced valves, is purchased at the expense of not being perfectly tight, and the advantage itself is unimportant with a construction of valves and cylinder such as we have adopted for our engines, and which we have already explained. Piston valves have also the decided disad-

vantage of increased clearance, which means that much more steam consumption, and this additional clearance over that found in engines with flat valves is a necessity from their mode of construction.

ROTARY PISTON VALVES.

Rotary piston valves are liable to the same objections, on the score of unequal and excessive wear, which have been urged against the piston valve; and, although various expedients have been adopted to lessen the injurious effect of this wear, and to render the valves and seats more readily re-fitted when leakage occurs, it will hardly ever be possible to overcome this serious defect. The direction of improvement in steam engine construction is now tending to the adoption of flat valves, simple in form and of such size and travel that the small amount of power required to operate them renders special means of balancing an unnecessary and useless complication.

RIGHT AND LEFT HAND ENGINES.

The cuts Figs. 3, 4 and 5 are those of right hand engines, in Fig. 5, which is a plan of a Class C engine, it will be observed that when we stand at the cylinder end and look towards the fly-wheel, the frame, at that part beyond the guides, curves to the right, and, that the main bearing, shaft and fly-wheel are on the right hand side of a line drawn to represent the axis of the cylinder A left hand engine will have a bed or girder which curves to the left and the fly-wheel etc. will be on the left hand side of a centre line through the cylinder.

ENGINES RUNNING OVER AND UNDER,

Standing as before at the cylinder end and facing the crank end, if the top of the fly-wheel revolves from the observer, the engine "runs over," and, if it revolves towards him, the engine "runs under;" this applies to either right or left hand engines.

IMPROVED METALLIC PACKING FOR PISTON-RODS, ETC.

Fig. 32 illustrates our improved metallic packing, the plan and sections at the lower part of the figure show one ring which is seen to consist of eight segments, the inner circle being of the diameter of the valve-stem. The sectioned portion of the figure beyond the inner circle represents brass and the lighter shaded portion babbitt metal, The four alternate segments, it will be observed, form a continuous ring of babbitt around the valve-stem or piston-rod and the babbitt is backed by what, except

for the small holes necessary to hold the metal in place, is a solid wall of brass. The large figure shows another section of the packing ring, the babbitt it will be seen does not extend the whole width of the ring, but there is a thin layer of metal left at the bottom of each segment upon which the babbitt rests. Referring again to the small sectional plan it is shown that the other four segments are solid brass and that the eight segments taken together form one ring. To hold the separate parts in place and to ensure a firm contact with the rod, the ring is enclosed by a corrugated or other suitable spring, the corrugated spring shown at the upper right hand part of the figure has eight flutes which bear each upon its own segment and thus presses them all against the rod. We use two or more of these rings to make up the required packing and place the rings so that the segments break joint. In the large figure

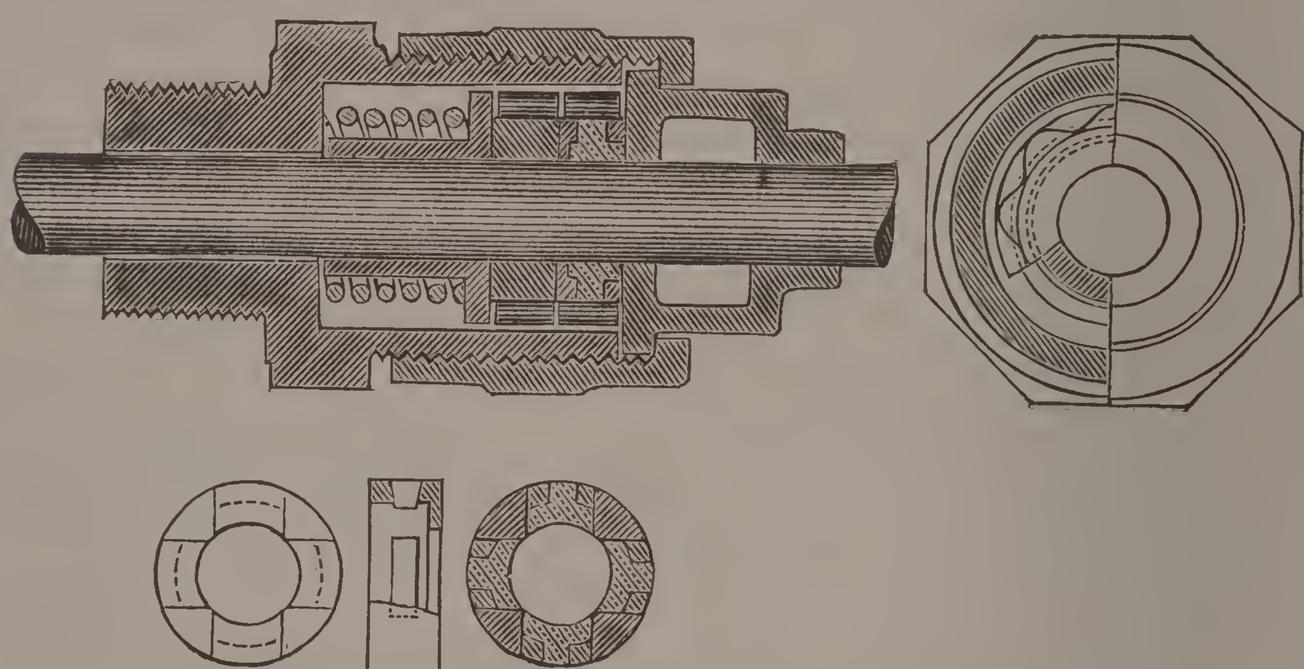


Fig. 32.

there are two of these rings within the stuffing box. They rest at their lower part upon a bushing which is surrounded by a spiral spring; at the outside end there is shown a hollow cap held in place by a nut bearing against its flange, this cap being hollow, any steam or water which might possible leak through is led away by a small pipe and the stuffing box and valve stem are thus kept clean and dry. The object of the spiral spring is to furnish an elastic backing for the rings, so that while they are held firmly enough to keep their position, they are not pressed together so much that the friction prevents the segments moving inward freely so as to bear against the rod and form a steam tight joint. These figures are intended merely to illustrate the principle of this packing which we use for piston-rods and valve-stems; changes in the details of arrangement are of course made to suit our different engines.

POINTS TO BE CONSIDERED WHEN SENDING ORDERS.

In the following articles it is intended to supply information and to offer suggestions to persons ordering engines with a complete outfit, so as to guide them in their selection and to facilitate the execution of orders.

CLASS OF ENGINE AND HORSE POWER NEEDED.

The first thing to be stated is what class of engine is desired and what horse power is required; this being determined upon, a suitable engine may be selected from among those given in the various tables of our several classes.

For best economy smooth working and close regulation of speed, an automatic engine should always be selected. For light powers, high speed and best economy the class A automatic engine is recommended; where less speed and high economy is desired one of our class B special automatic engines is recommended; for moderate speeds and high economy our class C standard automatic engines will be found most satisfactory; and a moderate speed with good economy, which comes next to that given by an automatic engine, may be obtained from a class D engine with fixed cut-off.

We have given tables under classes A, B & C in which the horse powers are based upon 80, 90 and 100 pounds boiler pressure, any one of which may be used according to preference.

But we recommend, in the best interests of our customers, the moderately high pressure of 90 pounds which, without requiring unusual strength for the boiler and engine, will be found to yield excellent economical results. For non-condensing engines we have found a cut-off at $\frac{1}{4}$ stroke to be the most economical point; our horse power ratings are based upon this cut-off, and a rate of revolution given in the tables such as is considered best for each engine. We would advise that our ratings, as given in the $\frac{1}{4}$ stroke column of the tables for 90 pounds pressure, be accepted as comprising the most favorable conditions. Condensing engines, however, will work economically at an earlier cut off than $\frac{1}{4}$, say $\frac{1}{5}$ or $\frac{1}{6}$ stroke.

PROVIDING FOR INCREASE OF POWER.

An engine of a size just suited to its work will, in general, be the most economical one to use, but frequently it is thought best to put in a somewhat larger engine than is immediately necessary in order to provide for future increase of power. We may increase or decrease the power of any one of our engines in several ways; they are rated at a certain speed, but they may he run 10 to 15 % faster or slower, we may also vary the point of cut-off or the steam pressure. Thus an engine rather larger than necessary may be run at a slower speed or may be made to cut-off earlier, say at $\frac{1}{5}$ stroke instead of at $\frac{1}{4}$ stroke, or we can preserve the same number of revolutions and $\frac{1}{4}$ stroke cut-off and adopt a modified steam pressure as explained in the article STEAM PRESSURE, page 7. Standard speed, a cut-off at $\frac{1}{4}$ stroke or 90 pounds pressure may be restored whenever more power is needed. A further increase of power may be had by using a condenser which, when properly applied to an engine, working with a fairly economical load, will add about 25% more power and is a most excellent and economical method. Beyond this, additional power is to be had by putting in an-

other engine, it is often advisable to bear this in mind at the start and so arrange the engine room that a second engine may be placed beside the first one and coupled on to the same shaft, or otherwise applied, which gives a pair of engines and makes a very good arrangement.

RIGHT AND LEFT HAND ENGINES, FLY-WHEELS, COUPLINGS, &c.

Having decided upon which size and grade of engine to use and whether it is to be condensing or non condensing, it is then necessary to state whether it is to be right hand or left hand and whether it is to be run over or under. Diagrams illustrating these last points have been already given and may be referred to in this connection. We desire also to know whether a band wheel is to be used or a fly wheel, if the latter, and the main shaft is to be coupled to the line shaft, whether we are to furnish the half coupling required on the main shaft or couplings for both main and line shaft. In this case it will be necessary to send the exact size of the line shaft and a better fit is ensured if we make both couplings. Couplings are not included in the outfit furnished with an engine; they are only occasionally employed when specially ordered and an additional charge is made for them. In some cases an engine is made with a fly-wheel and a pulley of much smaller diameter drives the belt. We will furnish such an arrangement whenever the conditions require it, the cost of a pulley being of course extra.

SELECTION OF A BOILER.

It will assist greatly in determining what kind of boiler to select, and to know how to settle upon a proper condenser or heater, in case either of these are to be employed, to have full information about the water supply. We should like to know whether there is at all times a plentiful supply, and also exact information about the quality, whether it is salt or fresh water, hard or soft water, or muddy and mingled with much gritty substance, or whether there is anything in solution likely to corrode and injure metallic surfaces Estimates of boilers suited to any engine will be furnished upon application, and our customers will be assisted in their own selection by reference to that part of our catalogue treating of boilers. When estimates are requested, be particular to state exactly what kind of a boiler is desired, and in cases where we are asked to suggest a suitable boiler, let us know something about the kind and quality of fuel to be used, with its cost. Where fuel is expensive it would be well to select a tubular boiler to ensure greater economy, but such a boiler might be forbidden in consequence of having water which formed scale whose frequent removal, so necessary to efficiency, could not be so well accomplished in a tubular boiler as in one of more simple construction. We shall always be ready to advise and would much prefer to select ourselves, a boiler suitable for an engine of given size, so as to secure reason-

able economy in fuel and adapted for whatever quality of water is available. This latter is a very important point, for where water is liable to deposit foreign matter and to form scale, the efficiency of a boiler becomes seriously impaired, unless ready means are provided for frequent examination and cleaning, therefore the construction of a boiler must be such as to admit of this. Some boilers, as plain cylinder and also flue boilers, afford ready access to every part of the interior, and a flue boiler, if used in cases where scale was likely to form, would be much more satisfactory than a tubular boiler.

CHIMNEYS.

For small engines an iron stack with plain breeching is commonly employed, and we furnish these unless otherwise ordered. Large engines should generally use a well proportioned brick chimney or stack and it is necessary to state with an order what kind of stack has been adopted. In order to properly locate the stack, as well as to fix other essential dimensions, we would always like to have sent a sketch of the engine and boiler room with dimensions marked on it, showing the desired position of engine and boilers and location of stack with reference to the boilers, dimensions to be measured from fixed lines as the walls of building ; also give the height from bottom of ash pit to underside of opening in chimney for breeching, and the height in clear of engine and boiler room. We should also desire to have located the position of well or other source of water supply and the depth of well from surface of ground to the lowest water level ever reached. We shall then be in possession of all the data necessary to cut all our pipes, etc., to right length and to make a working drawing, giving everything in proper position.

FEED PUMPS.

We prefer to use a pump for boiler feeding, because in many cases we have to furnish lime extracting heaters, and frequently employ condensers. Our engines, too, are sent to all parts of the country, meeting with every variety of water and with different degrees of skill on the part of engineers ; so that a pump, which is so easily understood and kept in order, answers our purpose in all cases better than any other kind of boiler feeder. Feed Pumps are included in a regular outfit, and they are always of ample capacity with a liberal margin for emergencies.

FEED-WATER HEATERS.

With non-condensing engines a heater conduces to economy since it makes use of heat contained in the exhaust steam, which otherwise would be wasted, to raise the temperature of the feed water. A heater also acts

as a purifier, in many cases where the water contains salts of lime, magnesia, &c. in solution they may be precipitated by boiling and so removed before reaching the boiler, thus preventing to considerable extent the formation of scale. Where water is hard and contains many impurities, a feed-water heater is an almost indispensable adjunct to a boiler. A good feed water heater will deliver water to a boiler at a temperature somewhere near 212°, ordinary feed-water being say 60°; this represents a saving of 152 units of heat or about 13 per cent. of the total heat of steam. There is also somewhat of a reduction in back pressure in the engine, the heater acting like a condenser in this respect, though only to a very limited extent. Considerable advantage results to the boilers when a heater is employed, because the feed-water is hot and does not injure the boiler by causing unequal contraction such as occurs when feeding cold water; but more especially a heater is designed to precipitate, by boiling, various impurities, which if allowed to enter the boiler, would form incrustations and deposits and thus seriously endanger its safety as well as reduce the efficiency by a large amount. We furnish a heater simple in construction and designed to meet all requirements. Its use is recommended as a valuable addition to a non-condensing engine and especially where there is impure water.

We do not wish to be understood, however, that a heater is only to be used with a non-condensing engine, for with condensing engines also we recommend that a heater shall always be placed between the engine and condenser, and where the exhaust pipes and heater are made in good proportion to the engine and the volume of steam that is to pass through them, no reduction whatever to the effective vacuum in the cylinder will result from this arrangement as condensing engines of our construction are in operation connected to the condenser with an intervening heater, which show twelve pounds of vacuum in the cylinder. So it will be seen that there is no loss whatever to charge against any gain that may result from the use of a heater in this way. The reason why we recommend this heater is, that while a good vacuum may be maintained by discharging the water from the condenser at a temperature of 100° to 110° or 115°, yet the temperature of the hot well is seldom more than 100° and is very often less. But the exhaust steam as it passes from the engine through the heater to the condenser has a temperature of 140° to 150°, and it follows that the temperature of the feed-water may be increased from the temperature of the hot well, which, as stated before, would be in general practice, about 100° or a little under this temperature, to a temperature of 140° to 150°.

When an independent condensing apparatus is used, it is advantageous to employ a second heater into which we would pass on its way to the boiler the feed-water from the first heater at its temperature of 140° to

150°. This heater is constructed to stand the pressure of the boiler and is otherwise constructed so as to adapt it for this peculiar duty.

We exhaust into this second heater the steam from the cylinder driving the air pump and exhaust steam from the steam pump or pumps used for boiler feeding or other purposes, or any other waste steam or heat that would otherwise be wasted. The feed-water absorbs this waste heat entirely, and it has occurred several times in our practice that we have in this way given to the feed-water before entering the boiler a temperature of 218°. The importance and economy of these temperatures as compared with a temperature of 100°, and the importance of any arrangement which while simple and inexpensive will give these results, need scarcely be discussed at greater length here.

The use of an auxiliary heater and the utilization of exhaust steam from the condenser steam cylinder, steam pumps for boiler feeding or other sources or waste heat and the drip water from steam jackets, is an arrangement original with us and entirely peculiar to our own practice. Small steam pumps as is well known are extremely wasteful of heat, and proper economy in the use of a steam jacket requires that the water which collects in the jacket shall be removed and the heat contained in it utilized, which, is all accomplished by the auxiliary heater in the most perfect way possible. This is manifest, when we consider that there is needed only a comparatively small amount of feed water to supply the necessary steam, and, the higher the temperature to which this small quantity of water can be raised by means of the high temperature exhaust steam, from the non-condensing steam pumps and heat from other sources which would otherwise be wasted heat, is all so much clear gain, it lessens the coal consumption, by just this amount and therefore an auxiliary heater, such as above described, which delivers water at a higher temperature than any other arrangement will effect a greater saving than is possible with any other kind of heater

THE INDICATOR.

The use of the indicator is now very general and its value is becoming more and more appreciated as an instrument which gives, in skilled hands, exact and valuable information upon various matters connected with the working of the steam engine which formerly were enveloped in mystery. Few high grade engines are now set up without having their valves adjusted for greatest efficiency as shown by diagrams taken with the indicator, nor are these engines accepted by the purchasers without having diagrams taken to show whether the steam is acting properly or not and to ascertain the horse-power which is developed by the engine, when running at its intended speed and under its proper load. When a man buys an engine he generally wants to know what it will cost to run it, there is a certain standard to which any engine may be referred in order to judge of its economy and this is the amount of coal consumed per hour for each horse-power developed. Many manufacturers while aware of what amount of coal is consumed, are totally ignorant of what power is being yielded by their engines and hence do not know whether they are working economically or not. They may be losing annually a large amount of money in consequence of having an engine which is wasteful of fuel, and it therefore becomes important to know just what a horse-power is costing and whether an engine of certain size is really developing that power which calculation shows it ought to be giving. Engines, designed with a special view to great economy, have been worked with an expenditure of two lbs. of coal per horse-power per hour and even less than two lbs.; but in general an engine may be considered as very good, if it yields a horse-power for every three lbs. of good coal consumed per hour; fuel of poorer quality will require perhaps 3½ to 4 lbs. which, bearing in mind the quality of coal, may still be considered a good performance. Engines in general, will consume various amounts of coal, other than these figures, sometimes running as high as 9 to 12 lbs. per horse-power per hour, which is extremely wasteful. An indicator diagram enables us to calculate the exact horse-power developed, and, knowing what coal is consumed, we can easily find how much is required per horse-power per hour and compare the figure found with figures which are considered to represent good economy. Large engines will, in general, be found much more economical than small engines, because although the sources of loss are the same, the proportion which they bear to the total power is very much less. But it must be remembered that the standard for efficiency referred to, includes the working of both engine and boiler, and that, to produce the best results, each must be designed to secure the highest possible economy. Sometimes a good economical engine is supplied with steam from boilers, whose evaporative efficiency is very low and, in such a case, it is not fair to charge the engine with a

defect which properly belongs to the boilers. In such a case, there can be made a separate test of the boilers, and the horse-power developed by the engine, compared with the amount of coal which would be required by a boiler of good evaporative power, say 8 or 9 lbs. of water per lb. of coal. The comparison will show the efficiency of the engine when used in connection with a proper boiler and is the only sure way to judge of the economy of the engine itself.

A steam engine indicator is an instrument used to draw a diagram which shows, upon a reduced scale, the motion of the piston and the pressure acting upon it at each point of its stroke. An indicator consists essentially of a small steam cylinder and a small drum upon which is rolled the paper for taking the diagram. The cylinder is provided with a piston whose motion is resisted by a spiral spring. Steam may be admitted beneath this piston and cause it to rise, or a vacuum created beneath it and cause it to fall, the amount of movement being a measure of the pressure, as in a spring balance. Motion from the piston is conveyed by a series of levers to a pencil, which is made to press against a slip of paper rolled upon the drum. When the instrument is in use, its cylinder is connected to either end of the large cylinder of the engine, and the drum is made by suitable means to revolve back and forth, having a motion which corresponds to that of the engine piston, only it is on a much reduced scale. Until steam is admitted to the indicator there is no pressure upon its piston, and if the pencil point is then pressed against the paper on the drum, it will, as the latter moves back and forth, trace a straight line, which is the line of atmospheric pressure. When steam is allowed to enter, the indicator piston rises against the resistance of the spring to a height corresponding to the steam pressure, and if this pressure remains unchanged during a stroke, a straight line parallel to the atmospheric line will be traced; when release takes place the piston instantly falls and the pencil moves with it, and when a return stroke of the engine occurs, the pencil will trace a line corresponding to the back pressure against which the engine piston is moving. This gives an idea of the process of tracing a diagram when steam follows full stroke; when a cut-off is used, the pencil traces the same line as before until the cut-off valve closes, when, as the pressures fall, there is traced a curve which gives the pressure at each point of the forward motion according to the law for expansion of steam. The length of a diagram drawn in this way represents on a smaller scale the stroke of the engine, and the line traced by the pencil shows the pressures acting upon the piston. These pressures are measured by the movement of the spring contained in the indicator, an inch of movement, or an inch of height above the atmospheric line on the diagram, representing so many pounds pressure, according to the spring used; thus a 30 lb. spring would be

compressed, so as to give the pencil a movement of one inch for 30 lbs. steam pressure, and a 40 lb. spring, one inch for 40 lbs. pressure, and so on. Having, then, a scale, in which one inch is divided into 30 or 40 parts, or any other number of parts such as ordinarily used, we can readily measure any pressure directly from the diagram when once we know what scale or spring has been employed.

In order to make familiar the various lines upon an indicator diagram, and to give an idea of their characteristics, we have prepared an ideal diagram which shows the various lines to which certain names have been given ; these will now be stated and the lines themselves briefly explained. All that it is designed to do is to give such a description that any one not familiar with indicator diagrams may be assisted to analyze them so as to appreciate the points of a good diagram, to be able to con-

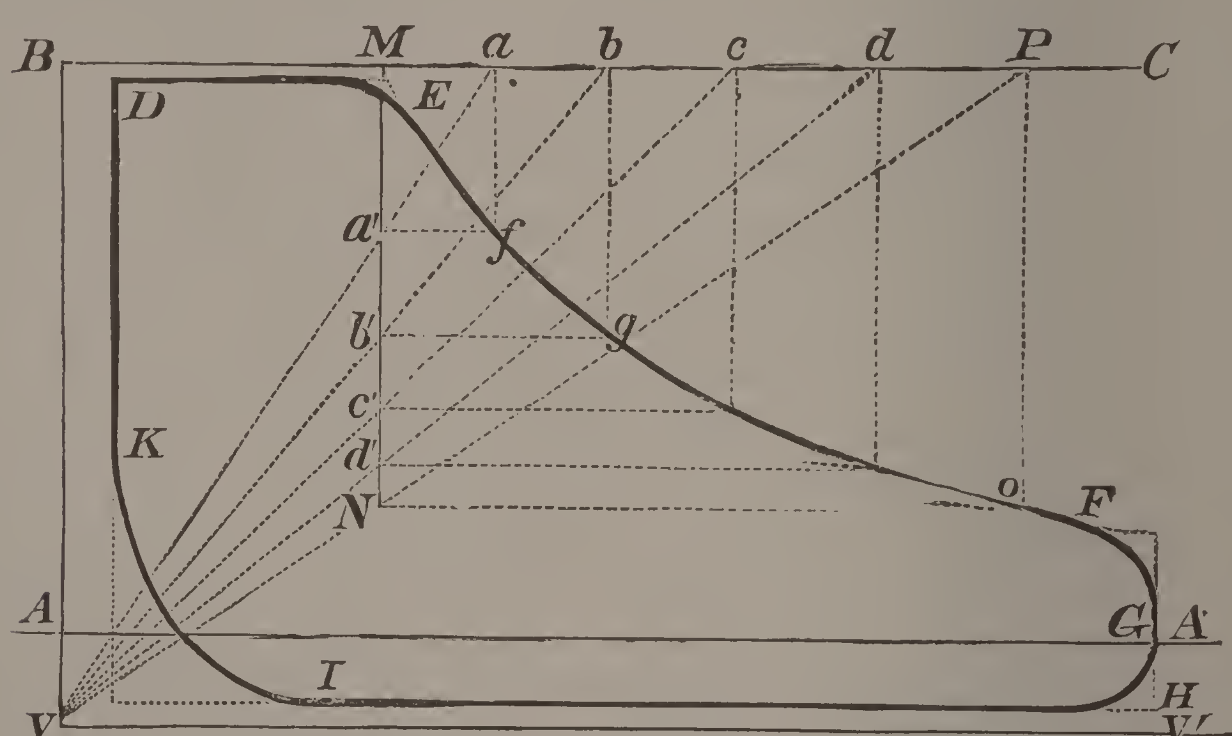


Fig. 33.

struct the theoretical curve for expansion, and to measure from a diagram the mean effective pressure and calculate the horse power of an engine from its diagram.

Fig. 33 is an ideal diagram showing each of the lines, supposing the action of the steam to be theoretically correct. In practice, the corners will be found more rounded and the expansion line will depart from the theoretical curve, but the nearer a diagram approaches the theoretical figure the better is the action.

The following names have been given to the lines of the diagram :

The atmospheric line A A.′ This line is drawn by the indicator before steam is admitted, when there is only atmospheric pressure upon the piston. From this line we measure pressures for non-condensing engines.

Line of perfect vacuum V V.′ This line is drawn parallel to the atmospheric line and at a distance of 14.7 lbs. below it, measured by the same scale as that for steam pressure. It must be remembered however, that 14.7 lbs. is the average pressure at the sea level and that the pressure becomes $\frac{1}{2}$ lb. less for each one thousand feet of elevation above this level. The vacuum line is that from which pressures are reckoned for condensing engines, and from which absolute pressures are taken, since there can be no lower pressure below it.

The clearance line B V is at right angles to A A,′ and at such a distance from K that the included space correctly represents the clearance in the engine, and since this additional quantity of steam must always be in the cylinder and passages and take part in the expansion which occurs after cut-off, it is necessary to draw this line and to add this space to the diagram, whenever the theoretical curve is constructed to compare with it the actual curve traced by the indicator.

Line of boiler pressure B C. This line is parallel to A A′ and represents the pressure in the boiler by gauge. It is needless to remark that all pressures must be laid off to the same scale as used with the indicator.

Admission line. K D represents this line; it should be parallel to A B, and when this is the case it shows that steam of full pressure is had at the commencement of a stroke.

The steam line D E should be parallel to B C and is invariably several pounds below it; the loss of pressure occurs from the steam being cooled after leaving the boiler, from friction in the pipes and bends, &c. and also in consequence of there being always required a difference of pressure in order to make the steam flow into the cylinder. The line represents the initial pressure acting upon the piston up to the point of cut-off and should be of unvarying height to show that full pressure is maintained.

Point of cut-off. This occurs at E. In the theoretical diagram the corner is abrupt, but in practice it is more or less rounded; when the valve is finally closed, the convex curve of the rounded corner changes to the concave curve of the expansion line, and the point of cut-off is properly located at the point where the direction of curvature changes.

Expansion line E F. The conditions most nearly realized in practice are such that when steam expands in an ordinary cylinder, its pressure falls in obedience to Mariotte's law, that is to say, the pressures are inversely as the volumes, and the curve which expresses the pressure for every point of the stroke is an equilateral hyperbola. This curve is easily constructed either directly from the calculated pressures or by a geometrical method which we will presently give.

Point of release. At F release takes place, the exact point being noted where the curve changes its direction.

Exhaust line F G, exhaust must occur early enough to allow the steam to get well out of the cylinder before a return stroke commences but yet no earlier than necessary because the exhaust line then falls so much below the theoretical expansion line that considerable loss of power occurs.

Line of counter pressure H I. In a non-condensing engine this line is usually one or more pounds above the atmospheric line and in condensing engines is 11 or 12 pounds below it, according to the vacuum in the condenser.

Compression line I K. When the exhaust valve closes at I, steam remaining in the cylinder is compressed and its pressure rises in proportion to the amount of the compression, which, with good cards in practice, is sufficient to raise the pressure of the confined steam to about one-half the initial pressure; the theoretical aim should be to have compression begin at such a point that steam in the clearance space is raised to full initial pressure at the commencement of each stroke, in this way the loss by clearance is to a great extent corrected. Compression is also useful to form an elastic cushion to gradually stop the piston at the end of a stroke, and by regulating it so that the steam is compressed to a suitable pressure, there is no shock from the entering steam when a new stroke begins; thus the proper regulation of the compression serves to make an engine work easily and smoothly as well as to prevent waste from clearance.

Method of drawing the expansion curve. In Fig. 33 there is shown a neat construction of the theoretical expansion curve, which should always be drawn upon the diagram in order to compare it with the actual line traced by the indicator. To make the construction it is necessary to know the clearance space so as to draw the clearance line B V from which expansion is reckoned, to draw B C the line of boiler pressure and also V V′, the line of perfect vacuum. Then take any point such as O, on the expansion line of the diagram; this point must not be later than F, the point of release, because here the exhaust line begins; from O draw O P at right angles to B C and O N at right angles to B V, join V and P and at N, where V P intersects O N, draw N M parallel to B V. Then M is the theoretical point of cut-off. The space M P can be divided into any number of parts which need not be equal, and lines drawn from V to these points a, b, c &c., cut the line M N in points a′, b′, c′ &c. From a and a′ are drawn lines parallel respectively to M N and O N and where they intersect is a point of the curve. The same operation for b and b′, gives another point and so on. When a little skill is acquired these lines need not be entirely drawn in, but only so much as to show the intersection which determines a point of the curve, and it is thus a very

easy and expeditious method for drawing the true curve upon an indicator diagram.

We will now define the terms initial, counter, terminal and mean effective pressure and explain how to calculate the indicated horse-power of an engine

Initial pressure, is that pressure which acts upon the piston at the commencement of a stroke up to the point of cut-off; it is always somewhat below boiler pressure. Counter pressure, or back pressure, is the pressure against which the piston moves; it acts as a resistance and has to be deducted from the pressure acting in front of the piston in order to give the pressure which is effective in producing motion. Terminal pressure, is that pressure which the steam would have supposing it to expand to the end of the stroke, instead of being released earlier. This pressure is found by extending the expansion curve to the end of the stroke, and measuring the height of the extremity of the curve above the vacuum line.

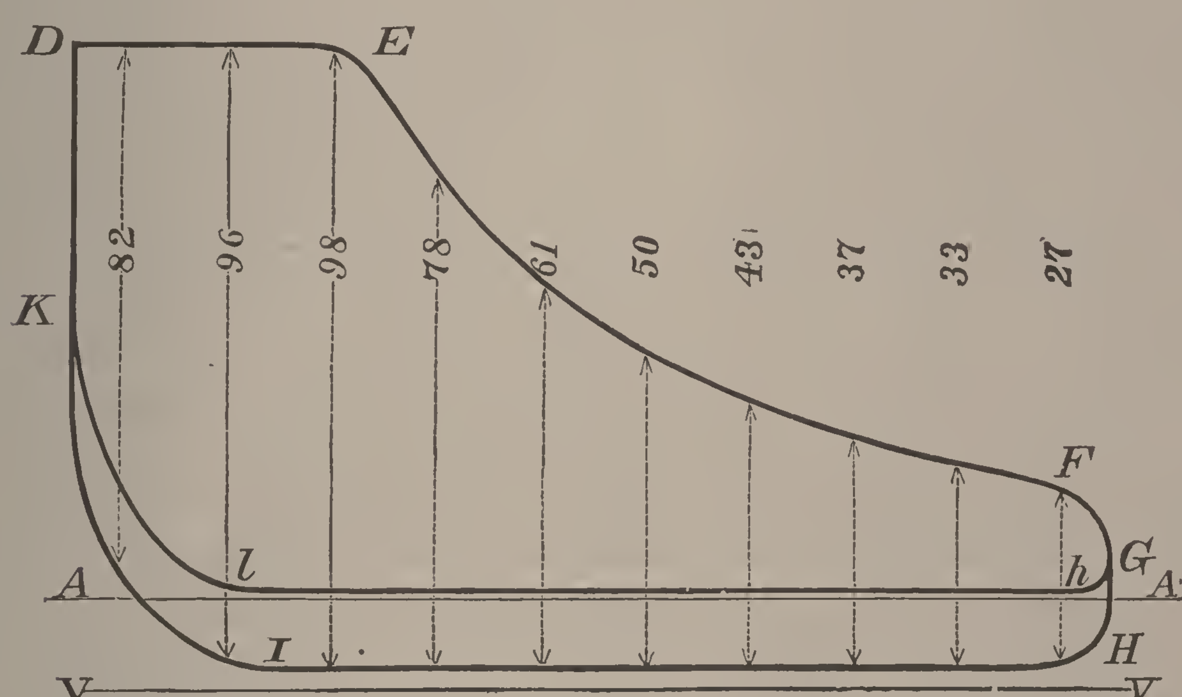


Fig. 34.

The mean effective pressure is the difference between the average steam pressure acting to propel the piston and the average counter pressure against which it moves. We may obtain this pressure directly from an indicator diagram, to do this we divide the length of the diagram into 10 equal spaces so taken that there is a half space at each end, 10 is a convenient number, but this is immaterial any other number may be used, the more numerous the spaces of course the greater the accuracy. Fig. 34 shows an ideal diagram, so divided by parallel lines at right angles to the atmospheric line. The figure D E F H I K represents a card from a condensing engine and the figure D E F h l K that from a non-condensing engine. The height of the upper part of the figure, measured on each of

these lines, counting from the vacuum line for a condensing engine, and from the atmospheric line for a non-condensing engine, shows the forward pressure acting upon the piston at each of these divisions. Should a non-condensing engine expand below the atmospheric pressure, then the pressures for that part of the curve below the atmospheric line are to be considered negative. Adding together all these pressures and substracting the sum of the back pressures measured from vacuum line for condensing engines and from the atmospheric line for non-condensing engines and dividing the result by ten or whatever number of equal spaces we have taken we get the mean effective pressure. But generally, it is not necessary to measure the steam pressures and counter pressures separately, we can take the proper scale and measure directly the length of ordinate included between the upper and lower lines of the diagram and dividing their sum by the number of intervals we get the mean effective pressure. Thus in Fig. 34 the sum of all the ordinates of effective pressure is, $82 + 96 + 98 + 78 + 61 + 50 + 43 + 37 + 33 + 27 = 605$. This divided by 10 gives 60.5 lbs. for the mean effective pressure. Where expansion below the atmosphere occurs with non-condensing engines, this rule is modified ; because, steam pressures below the atmospheric line not only cease to be effective but the piston moves against a back pressure from the atmosphere, these steam pressures are therefore considered negative, and, since the corresponding counter pressures are also negative pressures, it follows that the whole lengths of all the ordinates in the looped portion of the diagram are negative, hence they must be subtracted from the sum of the ordinates in the other part of the diagram and the remainder divided by the number of intervals gives the M. E. P. as before. Incidentally there is here shown the disadvantage of expanding below the atmosphere with non-condensing engines, for the back pressure being in excess of the direct pressure, the surplus simply acts as a drag on the fly wheel ; the direct pressure is thus worse than useless for producing motion. The same loss of power may occur with a condensing engine whenever, in consequence of insufficient vacuum, the line of back pressure rises above the lower part of the expansion line.

The method above shown is the one generally used to determine the M. E. P. but it is not to be considered as anything but a close approximation, it would be correct only when the number or intervals is indifinitely increased, so as to give the pressures at an infinite number of points. For purposes of calculation the theoretical M. E. P. is much more exact, and, where proper allowance is made for back pressure, it is a very useful and correct way of computing horse-powers. It will be convenient for this purpose to have a table of mean effective pressures for various steam pressures and various points of cut-off. Such a table we have calculated and will introduce it in this connection. The above explanations

will serve to give an idea of the lines of the diagram as they are ordinarily understood. Thus far only one diagram has been spoken of, but there should be a diagram taken from each end of the cylinder, these will be found to be similar, but not exactly alike, one reason is in consequence of the angularity of the connecting-rod and the presence of the piston-rod on only one side and the different compression required to give regular action, another is slight differences in setting the valves required by these and other causes. We have also taken it for granted in our explanation, that in a diagram, the lower line represents the back pressure acting upon the piston during a forward stroke. In reality, this is not the case, the real counter pressure against which the piston works is that represented by the lower line of a diagram taken from the other end of the cylinder. But since the M. E. P. should be obtained by averaging the pressures of both cards and deducting the back pressure in each, it will not affect the result, to consider the lower line in a diagram to mean the real counter pressure against which the piston is acting, even though such an assumption is not strictly correct. Having obtained the mean effective pressure either from the diagrams or from the table it is then easy to calculate the indicated horse-power of an engine. A horse-power is a conventional term, and expresses a rate of mechanical work, measured in foot-pounds for some unit of time as one second or one minute. 33,000 pounds raised one foot high in one minute is what is commonly understood to mean a horse power. To calculate the indicated horse-power of an engine we proceed thus:—The area of the piston in square inches multiplied by the mean effective pressure in pounds per square inch, multiplied by the piston speed in feet per minute (or by twice the stroke multiplied by the number of revolutions per minute) gives a certain number of foot-pounds of work, which divided by 33,000 gives the horse-power of the engine. Suppose for example that we have a 14x24 engine; piston speed 560 feet per minute, mean effective press- 38.65 lbs. per square inch, piston area 153.938 square inches. Then $153.938 \times 38.65 \times 560 \div 33{,}000 = 101$ horse-power. For the same size engine, and same speed, variations in power will occur as the mean effective pressure is varied; the other factors remaining as they were. This gives a means of facilitating calculations, for, if instead of 38.65 lbs., in the above calculation, we took a pressure of one pound then the result in horse-power would be 2.6121 horse-power; and in our tables this is what is called the horse-power constant for one pound M. E. P. If then we find the mean effective pressure and multiply it by the constant for the particular engine we get the horse-power at once, thus $2.6121 \times 38.65 = 101$ horse-power as before. When a different rate of revolution is taken, this constant is not correct, but it will be greater or less in proportion to the number of revolutions. Thus for 100 revolutions the constant should be

$2.6121 \div 140 \times 100$, that is divide the constant by the number of revolutions given in the table and multiply by the new rate of revolution, the result will be a new constant for that speed, which multiplied by the mean effective pressure gives the horse-power for the new speed. This constant will be found very convenient to ascertain what power will be developed with a given engine when the steam pressure, and therefore the M. E. P. is varied.

INDICATOR DIAGRAMS.

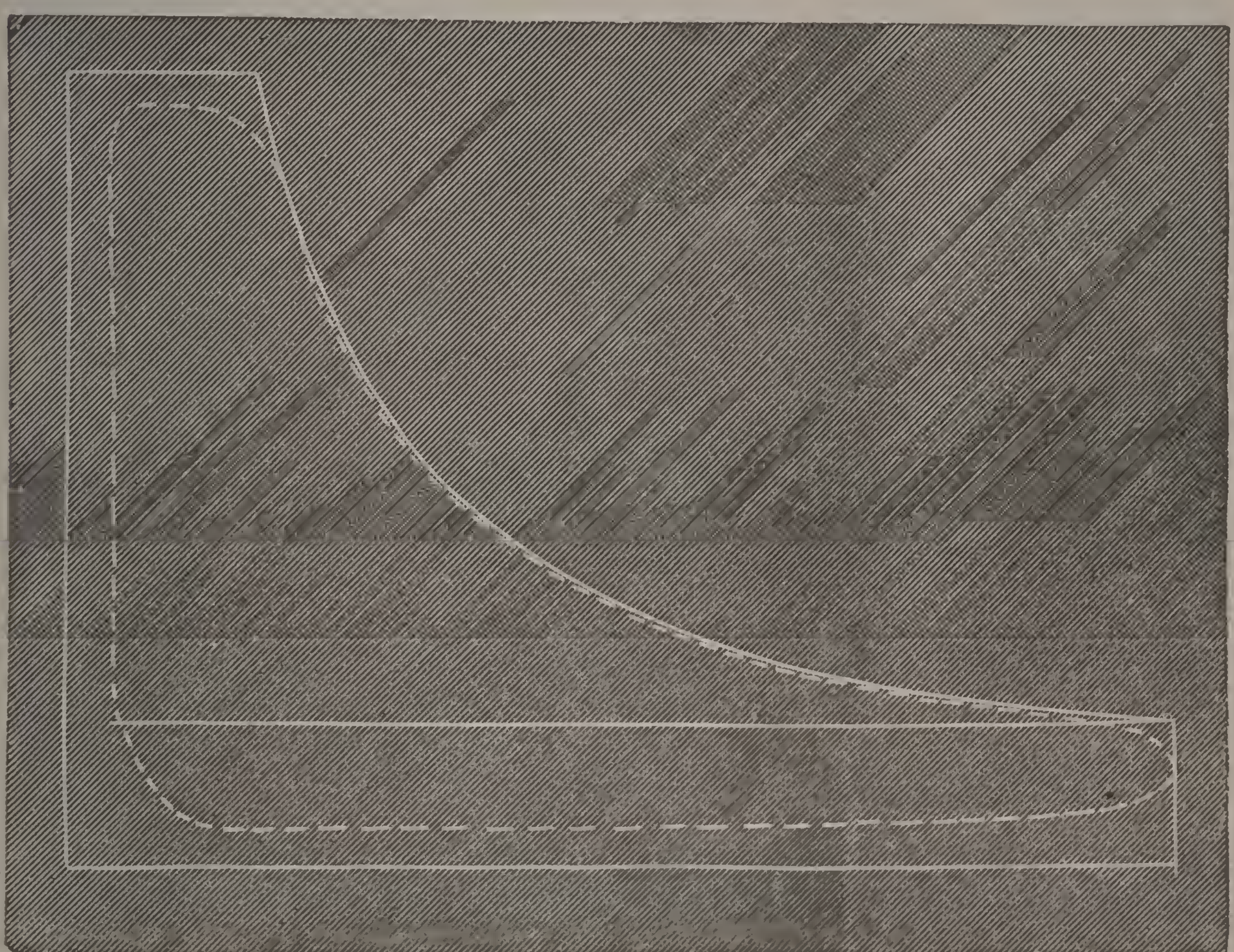
We now introduce a few diagrams, selected from among a large number which we have at hand; they are designed to show fairly the average working of our engine under ordinary conditions, and will be found to well illustrate what has previously been said about the theoretical points in good indicator diagrams, and to show how closely the peculiar excellence of our valve system permits the best results to be obtained.

The application of the indicator to an engine is really a scientific experiment, and should be undertaken with a degree of care and precaution against error, such as men of science adopt; the conclusions based upon the diagrams must also be judiciously made. While we are not of those who unduly exalt the office of the indicator to perform all sorts of impossible things, we do claim, and know, that when intelligently applied, the results are perfectly reliable, and are very valuable. There is no other way by which we may know positively that the valves have been properly set, and that the steam is acting to the best advantage, except by taking an indicator diagram from the engine, and correctly interpreting the various lines; nor, can we ascertain in any other way with a reasonable degree of certainty what power is being developed by the engine, or form any idea of how closely the actual power approaches the ordinary rating.

In these diagrams which follow, unless otherwise stated, the dotted portion is the actual figure traced by the indicator; and, the full line is the theoretical diagram of steam expanding from boiler pressure, with a cut-off equal to the given cut-off and clearance added, down to terminal pressure—pressures being measured from the atmospheric line in non-condensing engines, and from line of perfect vacuum in condensing engines.

Whenever the scale of the diagram is not given, it is to be understood that the original was reduced by photography, in order to make the wood cut of a suitable size for our pages; this reduction does not, of course, effect the truth of the diagram, since everything preserves the same proportion as before, but the diagram cannot be measured by the usual scale.

No. 1, scale 30 lbs., was taken from a 20x36 condensing engine, revolutions 73, steam pressure in boiler 65 pounds. The load on this engine is too light for economy, but the diagram is a good one; the admission line and steam line are good; the expansion line coincides very closely with the theoretical curve, and there is a free exhaust and excellent line of counter pressure. The compression might begin a little earlier with advantage.



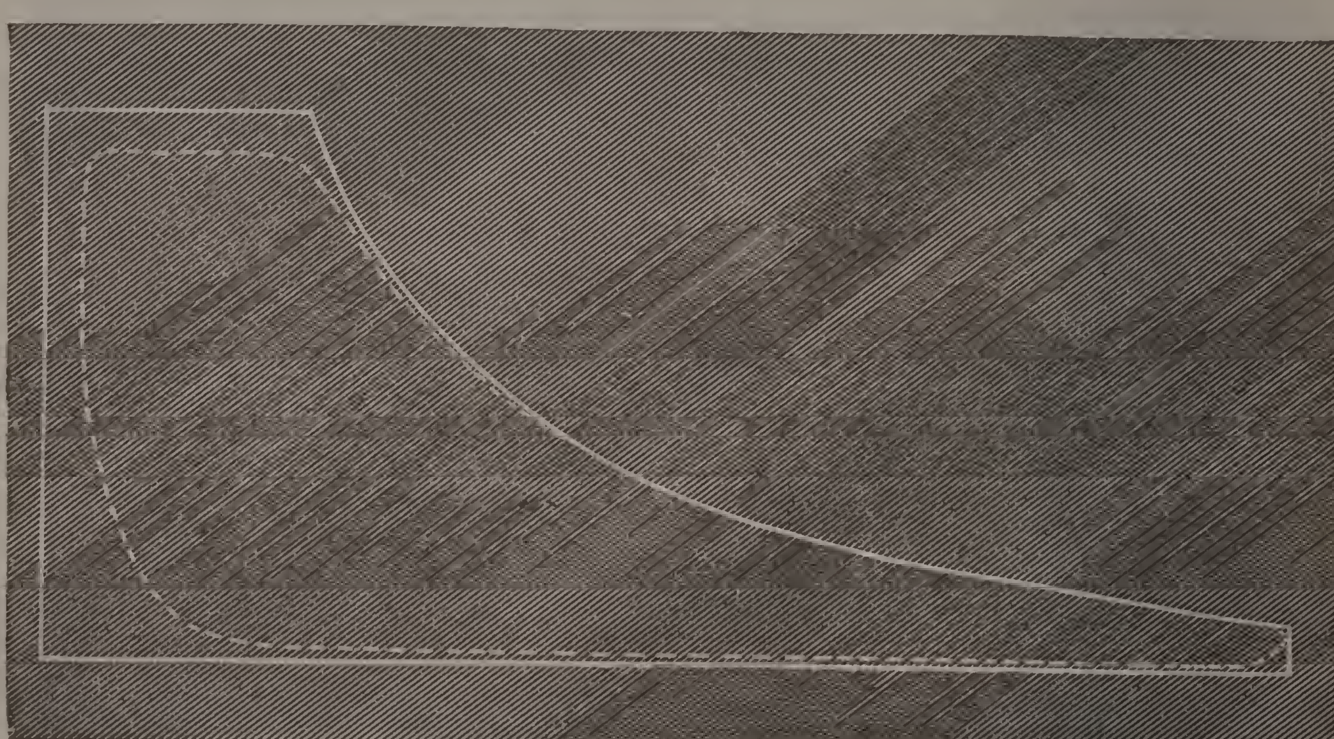
No. 1.

No. 2, scale 30 lbs., is a diagram from a 22x42 automatic condensing engine, revolutions 75, steam pressure 60 pounds; this is also taken with a light load. The point of cut off is well defined, and expansion and exhaust lines are good; the line of counter pressure runs nearly parallel with the line of perfect vacuum, and about four pounds above it. Here, we have a better compression than in No. 1.

No. 3. This diagram is from a 16x36 non-condensing engine; revolutions 90; steam pressure 90 lbs. The load on this engine is such as we consider a good one for ordinary economical working; the point of cut-off is at about one-fourth stroke. There is a good steam line parallel to that for boiler pressure, and only a few pounds below it. At the point of cut-off the corner is but slightly rounded, and the expansion curve fol-

No. 2.

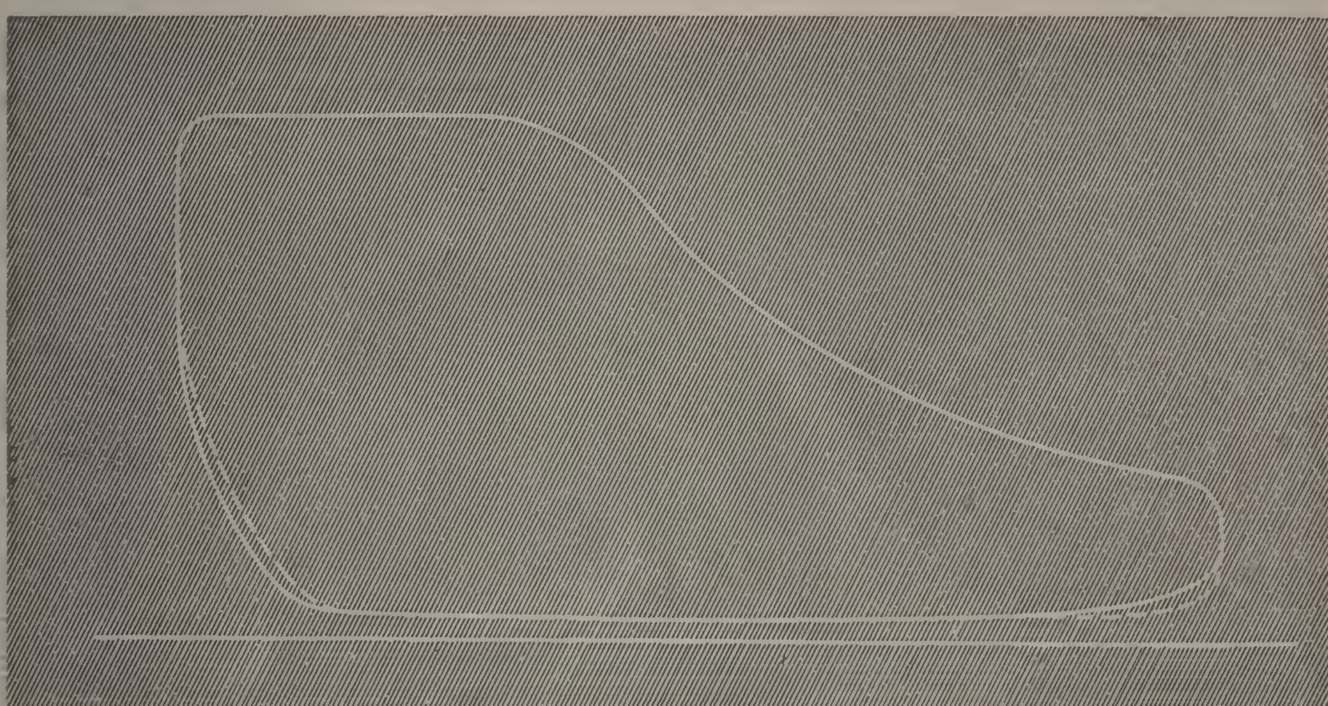
lows closely the theoretical line. The exhaust is excellent, as is also the line for back pressure, which comes close to the atmospheric line, and there is a good compression line



No. 3.

Our arrangement of separate exhaust valves is such that we can always set them to secure any desired release or compression. It is important to have a free exhaust without allowing release to take place too early ;

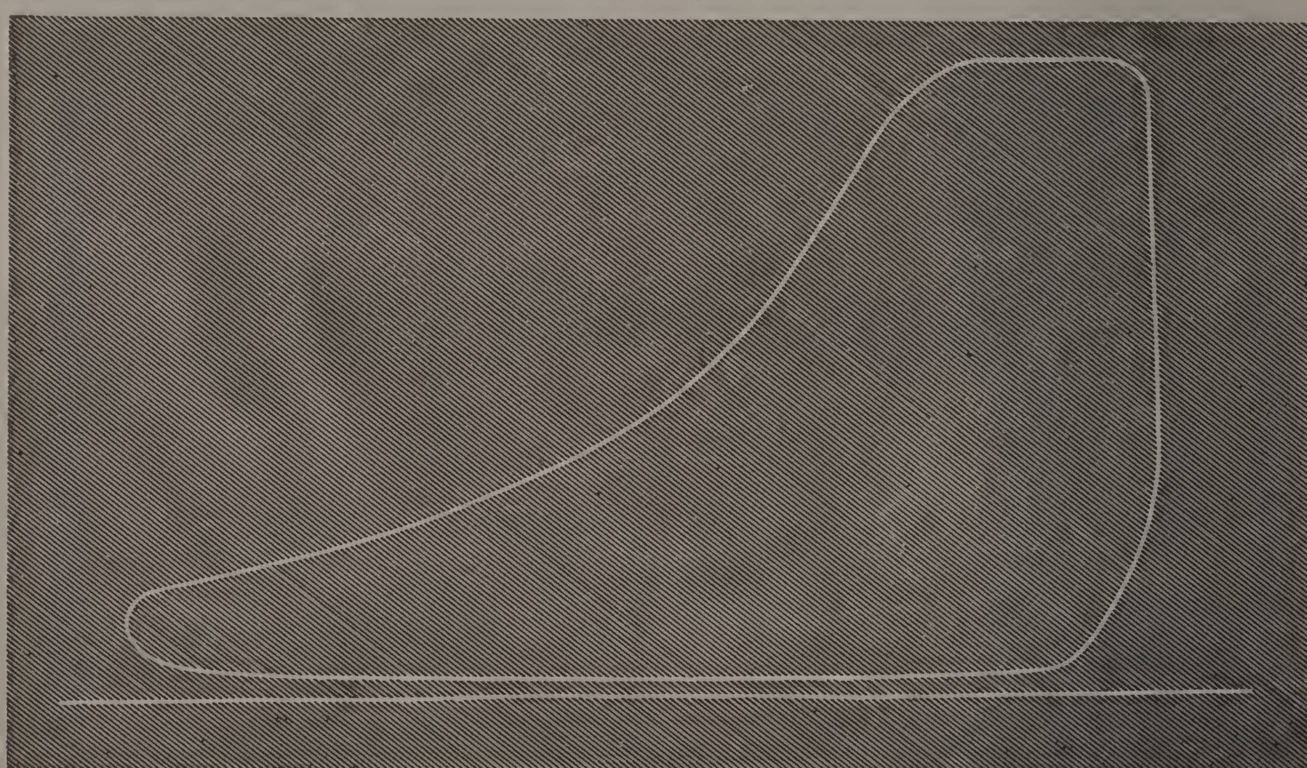
at the same time compression must be arranged so that steam in the clearance space may be compressed to proper pressure at the beginning of each stroke. Owing to the much lower counter pressure in condensing engines, it is more difficult to secure suitable compression than with non-condensing engines ; we have to commence to cushion earlier, and wish to do so without having to exhaust any earlier. In most other engines an early compression means an early release, but in our engine these two points can be adjusted independently, as we shall now show. When a valve motion is designed, we always give sufficient travel, which, while small, still allows a certain range for adjustment, and, in consequence of having valves at each end of the cylinder, which are independent of each other, we have practically an adjustable amount of lap. Suppose that we have the steam admission and exhaust release arranged



No. 4.

to our satisfaction, but that it is desired to give more compression, instead of deranging two things to effect an improvement in one point, we can bring them all to be just as we desire. Thus we can shift our main eccentric so as to give an earlier closure to the exhaust valve for our desired compression, and give more lap, so as to preserve the same lead as before, and therefore the same release. Nor do we really disturb the main valve, because it may be moved so as to have more lap and keep the same lead or opening at the commencement of a stroke as before, while its earlier closure does not affect the working, because we have allowed ourselves a limit of movement for adjustment. It will be apparent how easily and accurately our valves may be set, the change may be made if a non-condensing engine is to be worked as a condensing engine and still run smoothly, or to secure such results as the indicator shows to be desirable.

To illustrate these points we have inserted diagrams from one of our 14x30 Class C engines. In No. 4 it will be seen that the release is not early enough, and that in consequence of this, the back pressure at the commencement of the return stroke is much too high. This shows the effect of an improper valve setting; to remedy this defect, the main eccentric should be advanced slightly; this gives the exhaust valve an earlier opening, and produces an exhaust line and line of back pressure, as shown by the dotted curved line at this portion of the figure; at the same time the compression begins a little earlier, as shown by the dotted line at the point of compression. To preserve the same linear lead for the main valves, as before, we then spread them so as to give enough lap to compensate for the angular advance. This causes an earlier cut-off for the main valve, but not enough earlier than is within the limits assigned to the cut-off valve.



No. 5.

If we had wished to have an earlier release, and retain the same compression line, this could be accomplished by moving the exhaust valves towards each other, the same distance that the main steam valves had been spread, or moved apart from each other.

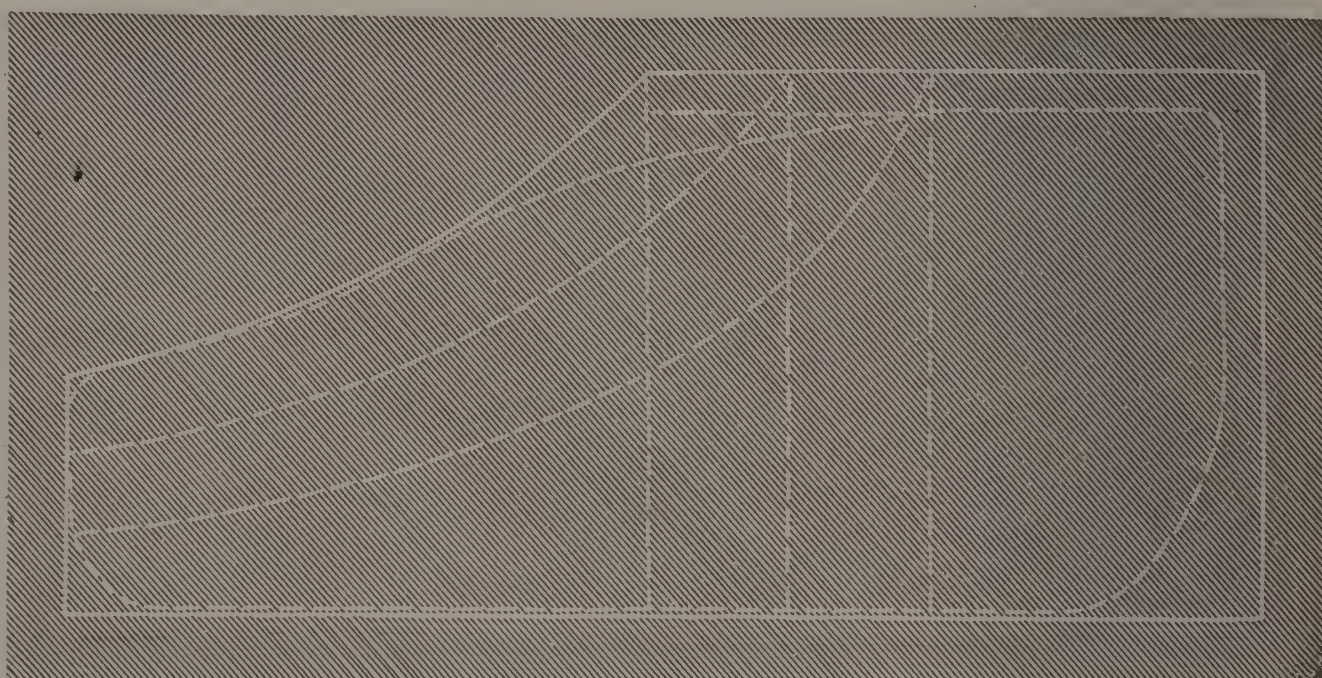
The line of back pressure in the diagram is nearly four pounds above the atmospheric line, whereas it should not be more than half a pound above this line, as shown in No. 6. This defect was in consequence of an improper connection with the heater, and the diagram, in contrast to No. 6, shows how a good engine may be given imperfect working, if those who attend to the erecting and valve setting do not perform their work properly, and do not make use of the indicator to correct any error or oversight that may occur. The two diagrams, No. 4 and No. 5 from the 14x30 engine show the benefit to be

derived from high pressure steam and expansion at an economical rate, compared with steam of lower pressure allowed to follow further. In No. 4, scale 50 lbs. the boiler pressure is about 85 lbs.; the point of cut-off about $\frac{3}{8}$ stroke, and the M. E. P. is 51 lbs., which gives, at 112 revolutions, 133.2 horse-power. The terminal pressure 39.7 lbs. is high; this pressure, representing the steam consumed and the mean pressure above vacuum representing the gross work done in a stroke, if we divide the latter figure by the terminal pressure, we obtain a result which shows the gain by expansion: thus, $69.7 \div 39.7 = 1.75$, the gain by expansion with 85 lbs. steam cut-off at $\frac{3}{8}$ stroke. No. 5, scale 50 lbs., is a card from the same engine only using steam of 102 lbs., and cutting off at $\frac{1}{4}$, instead of at $\frac{3}{8}$; here the M. E. P. is 49.29 lbs., yielding, at 112 revolutions, 128.62 horse-power. We have a lower terminal pressure, which is 32.7 lbs. Dividing as before, the mean absolute pressure, by the terminal pressure, we get $67.99 \div 32.7 = 2.08$, the gain by expansion in this case.

This gain, compared with the former figure, and a comparison of the horse-powers calculated from the two cards, shows that with a cut-off at $\frac{1}{4}$ stroke, with steam at 102 lbs., we obtain within four per cent. of the power given by steam at 85 lbs. cut-off at $\frac{3}{8}$ stroke; and, that we thus secure practically the same power with about 17 % less steam consumption. The boiler used with this engine is 64″ diameter x16 feet long, with 50 tubes four inches in diameter; it is such a boiler as we furnish and consider to be proper for one of our 14x30 Class C engines. A comparison of the horse power developed by the engine with the power ratings in our tables will show that our engines and boilers may be fully relied upon, to yield an economical power beyond our ratings. It is to be remembered that in these diagrams the back pressure is high; this was a fault in the connection with the heater, as already explained, and was very easily remedied. But if the difficulty had not existed, the same steam, taken from the boiler, and which is indicated by the diagrams, would have yielded 10 additional effective horse-power, instead of being uselessly expended in overcoming back pressure; and, the duty that would have been economically obtained from this engine and boiler is, therefore, entitled to be increased from 128.6 horse-power to 138.6 horse-power. The boiler fired quite easily, and without crowding in any way, at the time these cards were taken. These two cards, illustrating so well several points of interest and importance, we have gone into the subject a little more thoroughly, so that our patrons may know that we have given these important and vital matters our careful, intelligent and practical consideration; and, also, to show that the power ratings given by us, both for our engines and boilers, are sustained by actual and solid results.

No. 6 is from an 18x36 Automatic Cummer engine, the cut-off is at $\frac{1}{2}$ stroke, boiler pressure 87 lbs. a higher pressure being used in winter, initial pressure 81 lbs., revolutions 100, horse-power developed 300 horse-power.

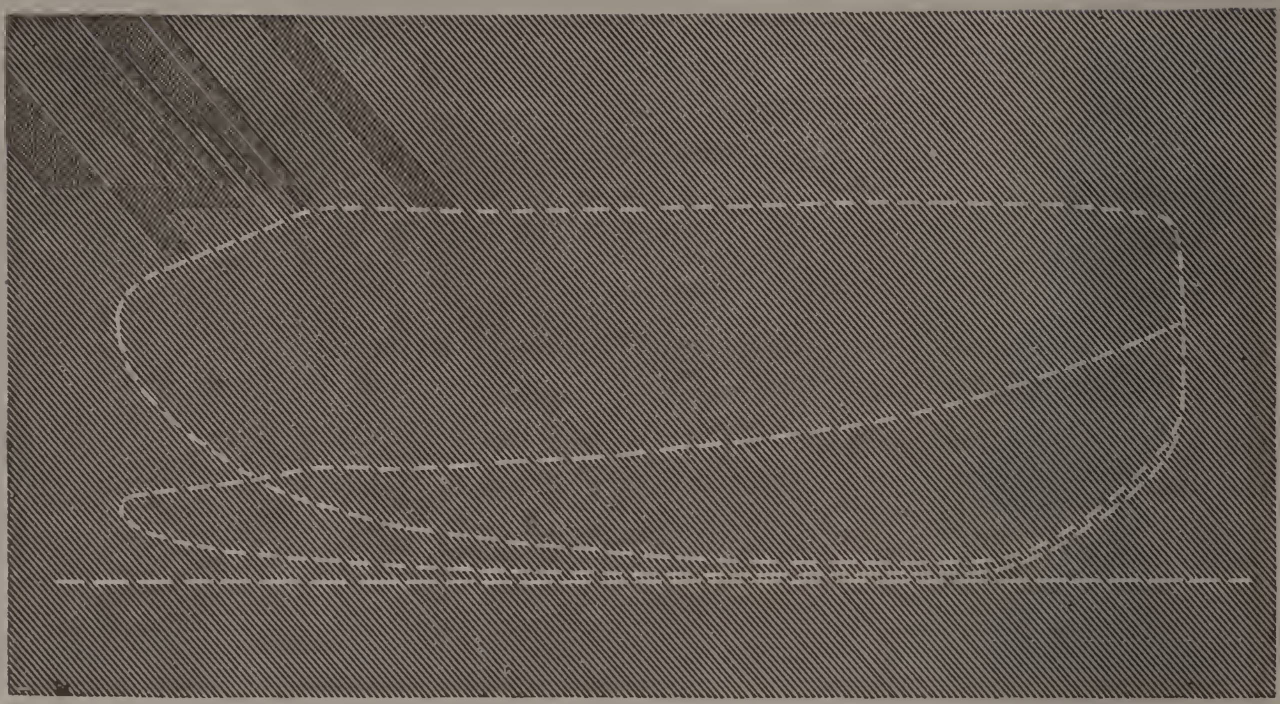
This engine is used to drive a saw mill. The full power required by the mill varies from 250 to 325 horse-power, according to the size of the timber and its condition, whether frosty or otherwise. From 60 to 65 horse-power is absorbed by the friction of the mill and the smaller saws, the latter consist of one gang edger, one gang bolter and one gang lath mill, and this is the only constant load upon the engine. The large 6 foot circular saw makes 650 revolutions per minute with a steam feed which advances the carriage $13\frac{1}{2}$ inches for each revolution of the saw. When in the cut, the large circular absorbs 150 to 250 horse-power which is in addition to the power required by the mill, and, the mo-



No. 6.

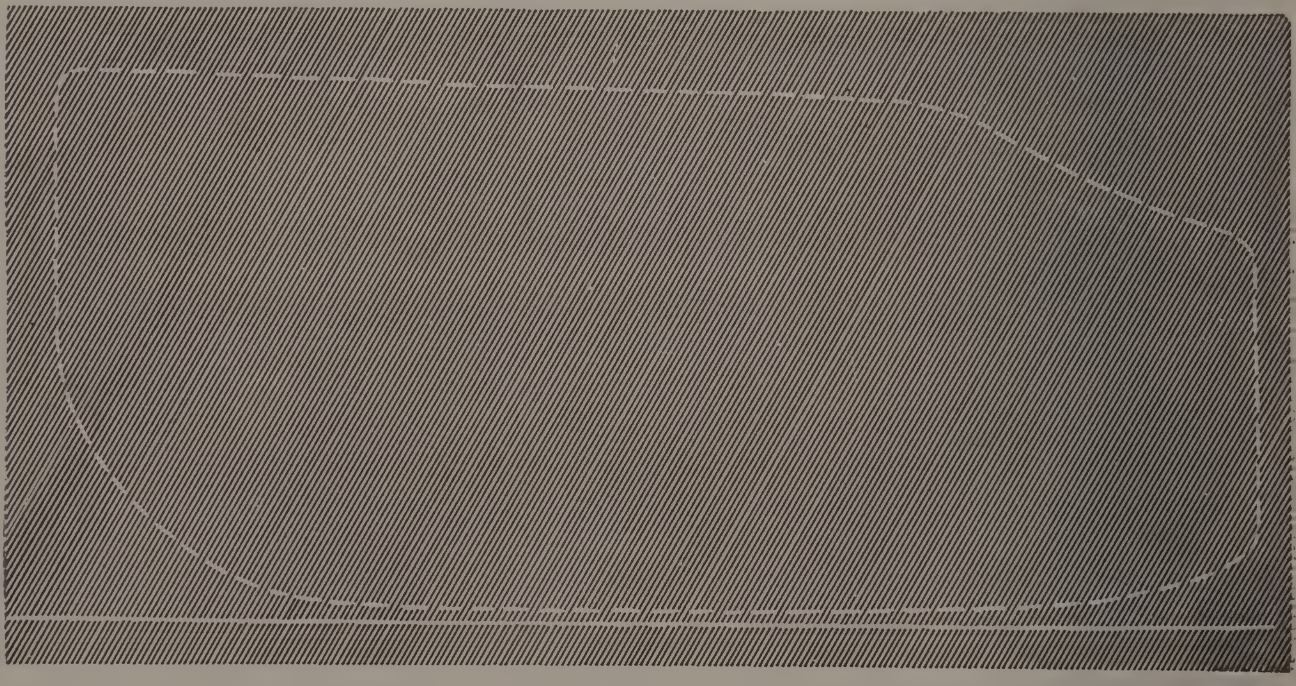
ment the saw is out of the cut the load is instantly reduced to the friction load of 60 to 65 horse-power; this sudden variation of power occurs several times per minute. There is an interval of time, however, when a new log is being put on the carriage, where the engine might race if not controlled by the governor, but the governor holds the engine so closely to its normal speed that there are no perceptible variations in its revolutions with any load that may vary from the friction load to that at half stroke; and from the friction load to $\frac{7}{8}$ stroke, the variations do not exceed four revolutions. Our expert and the owners of the engine assert that up to half stroke the variation was so slight that they could not count close enough to detect with certainty a variation of even one revolution from standard speed. This engine is working to maximum capacity and the above card which shows the action of the steam under a full load we consider to be an unusually good one. The steam line is

parallel to the atmospheric line up to $\frac{1}{4}$ stroke, beyond this there is shown a slight wire-drawing up to the point of cut-off at $\frac{1}{2}$ stroke. The loss of pressure from this cause is but very little, and is more apparent than real; it amounts to but $1\frac{1}{4}$ lbs for a whole stroke which is more than compensated for by other advantages as will appear later on. This card shows a good expansion line and a remarkably good exhaust line, the large quantity of steam in the cylinder is exhausted as quickly and freely as with a cut-off very early in the stroke. The line of counter pressure shows very little resistance and the compression line and admission line are both good. For the purpose of discussing some of the features of our valve system, we have drawn in dotted lines upon the diagram the theoretical curves for $\frac{1}{4}$ and $\frac{3}{8}$ cut-off, and, there is shown the completed figure for each of these points; referring now to the diagram it will be seen, that up to $\frac{1}{4}$ stroke there is no diminution in the height of the steam line and that at $\frac{3}{8}$ cut-off the loss of pressure is but very slight, amounting to only $\frac{1}{4}$ of a pound for a whole stroke. Our economical range for expansion is from $\frac{1}{5}$ to $\frac{3}{8}$ cut-off, the best economy being had for a cut-off at $\frac{1}{4}$ stroke; at this point, and also for $\frac{1}{5}$ and under, there is no loss of pressure whatever and at $\frac{3}{8}$ stroke it amounts in this case to only $\frac{1}{4}$ of a pound which is inconsiderable. For a cut-off at $\frac{1}{2}$ stroke there is the reduction in pressure such as shown by the diagram and for $\frac{7}{8}$ there would be somewhat more; now although we could make our ports and valves of such a size that there would be no wire-drawing whatever, even when following as far as $\frac{7}{8}$ stroke, yet increasing the size and travel of our valves would increase the friction very much and we have aimed to keep this as low as possible. So that it is a matter of free choice, and not of necessity, which has led us to submit to a little wire-drawing, after passing the economical range of expansion, in order to secure the great advantage of a plain, unbalanced valve moving with but very little friction, and, it will be seen, that in effect we sacrifice nothing, because our engines are designed to work economically with a cut-off varying from $\frac{1}{5}$ to $\frac{3}{8}$, and, within these limits, there is no appreciable loss. Beyond $\frac{3}{8}$ cut-off the engine should not be called upon to go for regular working, for, while a cut-off later than $\frac{3}{8}$ yields more power, it does not give enough expansion for good economy and an engine should only occasionally be called upon to exert its full capacity; it should be large enough to perform its regular work with a cut-off varying from $\frac{1}{5}$ to $\frac{3}{8}$. For an expansion within this range, our valves and port proportions are such, that our engines have been shown to yield an exceptionally high economy, placing them in the very formost rank of economical steam-users, and this without encumbering them with the unnecessary and injurious friction and wear inseparable from large valves and frictional surfaces. Our long and watchful experience enables us to decide upon these proportions without erring on either side as is demonstrated by our diagrams and the working of our engines.



No. 7.

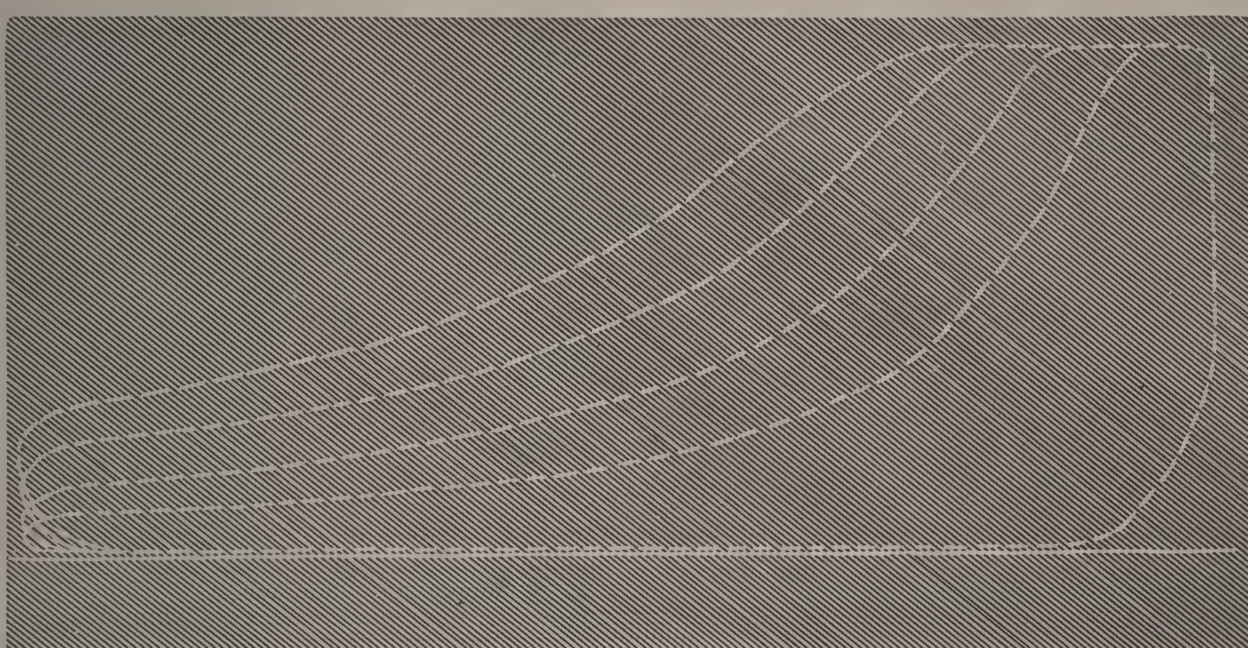
No. 7, scale 60 lbs. is a card from a 20x24 throttling engine, fitted with a plain slide valve cutting off at $\frac{7}{8}$ stroke; it may be considered a very good average card from an ordinary engine of this class. We introduce it here to show the improvement which may be effected when the valves and ports are given proper proportions. The boiler pressure used when this diagram was taken was 104 lbs., which is exceptionally high for this class of engine, the best initial pressure that could obtained was 71 lbs. representing a loss of 30 per cent., and the best mean effective pressure was 56 lbs. yielding 273 horse-power. We then made a change in the valves and ports, put on a better governor and produced a card such as No. 8; here the boiler pressure is 93 lbs. initial pressure 78 lbs., showing a loss by throttling of only 16 per cent. The



No. 8.

engine was made to cut-off at $\frac{3}{4}$ stroke which gives a mean effective pressure of 67.4 lbs., and a horse-power of 318.5 horse-power. This gives an idea of the great benefit which such j udicious changes will secure with an ordinary engine; much better results than this may be expected from our Class E plain slide valve engines which have been carefully designed throughout and correct proportions adopted for distributing the steam.

No. 9 is from the 18x36 engine before spoken of, steam pressure by guage 91 lbs., revolutions 100. The diagrams on this card show the



No. 9.

effect of a varying load; the approximate points of cut-off are $\frac{1}{10}$, $\frac{1}{6}$, $\frac{1}{5}$ and $\frac{1}{3}$ stroke; the cut-off at $\frac{1}{10}$ corresponds to about 60 horse-power, the frictional load; the other loads correspond to the different depths of cut made by the circular saw; it takes about three to six seconds, according to the condition of the timber, to go through a log 30 feet long, and all these changes including that for maximum capacity in No. 6 were made within 13 seconds. It only requires about one to two seconds for the governor to change the power developed from that required by the friction load to the full power at half stroke.

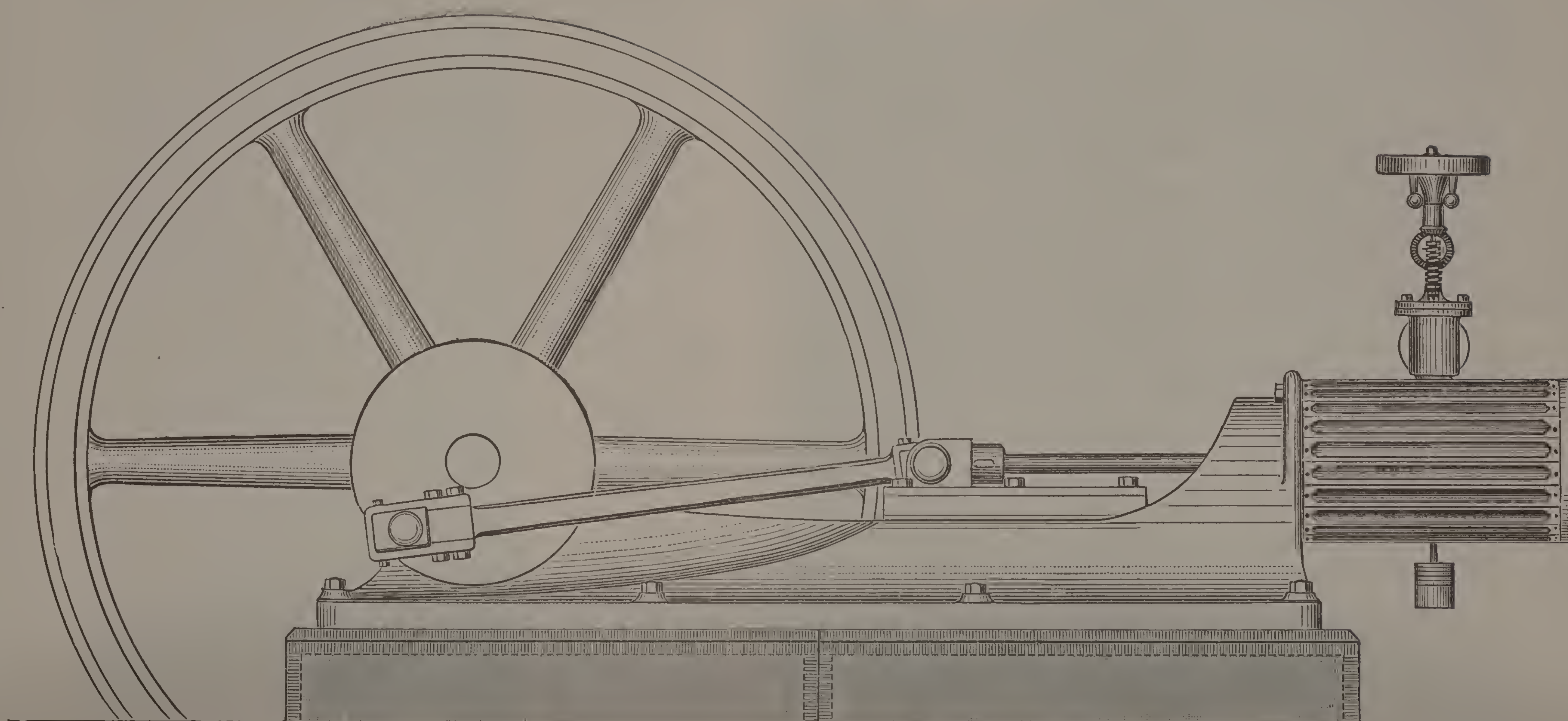


Fig. 35. Elevation of Engine. Class D.

CLASS D ENGINES.

The general design of this engine is seen in Fig. 35. As soon as cuts are prepared we shall illustrate and describe the various details, as we have already done with Class C; until then it will not be possible to make intelligible anything more than a general description. The bed plate, as will be seen, is all in one casting, and is supported for its entire length upon the foundation, while the cylinder overhangs and is attached to the frame in the same manner as is adopted with our Class B engines. It will be seen that the cross-head and guides have a different form from that in our engines previously described. For the cross-head, instead of having a bearing surface above and below the connecting-rod pin, the bearing surfaces are all placed below the pin. The lower part of the cross-head rests upon a broad, flat surface, and the cross-head is held in place by means of two flat shoes, one on either side, which form the upper guides; one of these shoes, with its bolts for adjustment, is shown in the elevation. The sliding surfaces of the cross-head are formed of anti-friction metal; there are no separate gibs, with means for adjustment like those used with our Class B and C engines; the shoes above mentioned give one means for taking up wear; and, when much worn, it is an easy and inexpensive matter to line up the cross-head, and renew the anti-friction metal surfaces. We use a disc crank for these engines, and the same form of connecting-rod as with classes B and C. The main and outboard bearings are also of the same general form as those for our automatic engines; the bottom of the bearing is anti-friction metal, and there are quarter boxes, with means for adjusting them, and also the cap, while a similar method of lubrication is provided as with those engines. For our Class D engines we use plain flat valves, and have devised a special form of valve so that, while one main valve admits steam to either end of the cylinder, and also exhausts the steam, the travel of the valve, and the surface exposed to steam pressure is small, and therefore such valves require but little power to move them; at the same time, by having an exhaust cavity at each end of the valve, there is, to all intents and purposes, a separate steam and exhaust valve for each end of the cylinder; thus the steam passages are made short and direct, keeping the clearance much lower than is usual in this form of engine. Upon the back of the main valve is another flat valve for cut-off; this valve is of such form that steam is admitted from two openings at once; a small movement thus suffices to give a large port opening in like manner as with our other valves already described. These engines have a cut-off which is fixed at $\frac{1}{4}$ or $\frac{3}{8}$ stroke, the eccentric and shaft have key-ways provided, so that either point of cut-off may be used as desired. The economy secured with these engines is second only to that

obtained with our automatic engines, while the design, workmanship and materials are equally as good. We may briefly point out the reason for the difference in economy between these two classes of engines; if it were possible to have such a thing as a constant load upon an engine, then, by adopting an economical point of cut-off, say $\frac{1}{4}$ stroke, and using steam of full boiler pressure, there would be little difference in economy in favor of either engine. But in point of fact, the load upon an engine in ordinary use is continually changing, and while an automatic engine measures off just that quantity of full pressure steam, which the load requires at each instant, the engine, with fixed cut-off, admits always the same volume of steam, and maintains standard speed by throttling, which varies the initial steam pressure. With automatic engines, we can use steam of nearly full boiler pressure, but with the other kind only about $\frac{4}{5}$ of this is available in the cylinder, as mentioned in the note under the tables for Class D; and, when engines are badly designed, the pressure is sometimes reduced to $\frac{2}{3}$, or even $\frac{1}{2}$ the boiler pressure. Engines, with fixed cut-off, are obliged to regulate by throttling or wire-drawing, and since any loss of boiler pressure is that much lost power, which an automatic engine is not subject to, except to a very slight extent, the reason for the superior economy of automatic engines is readily comprehended. These Class D engines, however, are well designed, and, when running under a reasonably constant load, will show an excellent economy, and are greatly superior to engines of the plain slide valve variety. In a former article, where a comparison was made between an automatic and a plain slide valve engine, much of the reasoning would apply with equal force to the cost and performance of an engine with fixed cut-off, as compared with an ordinary engine, and show a very decided economy in fuel in favor of the engine with fixed cut-off, as well as a cost of an outfit for a given power, which is only a slight advance upon the first cost of an ordinary engine with its outfit. The reason for this small difference in cost is, as before stated, in consequence of using steam of higher pressure, and getting as much power out of a small engine as was obtained with a plain slide valve engine of larger size; and, also, because there is greater economy in using steam which enables us to employ a smaller boiler. Thus we use for our Class D engines not over 12 square feet of heating surface to a horse-power, while ordinary plain slide valve engines will require 15 square feet for each horse power; this difference in the necessary boiler power, and the saving in first cost resulting from a smaller engine and smaller boilers, will often be sufficient to pay for the cut-off mechanism on an engine of this kind; and, the purchaser has thereafter all the advantage of the increased economy in fuel, which amounts to from 10 to 20 per cent.

In the design and construction of these Class D engines, we wish to call attention to the fact that they are proportioned throughout to develop such powers as are given in the tables, just as is the case with our automatic engines, and they may be relied upon to yield these powers. The frame, main bearing and crank, the connecting-rod, crank pin, cross-head, and all working parts are given such proportions that proper strength and stiffness is secured, as well as sufficient size given to ensure thorough lubrication and freedom from heating when developing the rated power. The ordinary plain slide valve engines cannot properly be compared in cost or construction with our Class D engines, of the same size cylinder, because they do not give nearly so much power, and their proportions will not allow them to be worked much beyond their ordinary ratings, while one of our Class D engines will give upwards of 50 % more power than the plain slide valve engine of the same size. In order to estimate what our Class D engines can do, it is necessary to refer both to the size of cylinder and the power they are able to develop. There are two ordinary ways by which some persons buy and sell engines; one is to estimate them at so much per horse-power, without reference to anything else; and the other is to base the cost simply upon the size of cylinder; but it is manifest that neither method is correct or satisfactory. It is not a good plan to buy an engine for so much per horse-power, because the engine may have to be run at such a high speed, and under so great a steam pressure, as to be unsafe, and to need constant care and watchfulness, as well as frequent repairs. It is equally unsatisfactory, and very misleading also, to judge of an engine and base its cost upon the size of cylinder alone, because, although the cylinder may have a sufficient capacity for developing a large power, yet, the engine itself may not be at all adapted in its working parts for anything beyond a much lower rating; and hence, it cannot justly be compared with an engine of the same size cylinder, but in which every part has been calculated for a power, such as the cylinder is able to yield. These facts should not be lost sight of by purchasers, and in estimating the capabilities of an engine, it should be determined whether a certain power can be developed with a certain sized cylinder, the engine working at a safe and moderate speed, and under a proper steam pressure; then it should be considered whether everything has been proportioned with reference to this power, in order that each part may have the necessary strength and stiffness, and that all the bearings and sliding surfaces are of sufficient size to prevent heating, and ensure thorough lubrication, After all these things have been considered, the question of economy, already discussed at the beginning of this article, will be found to be a very important matter. Each of these points, in turn, and in their relations to each other, have been carefully considered by us in the design and construction of these engines.

CLASS D. SIDE-VALVE ENGINES WITH FIXED CUT-OFF.

CYLINDER.		Revolutions per Minute.	Diameter Main Shaft, Inches.	BAND WHEEL.			BELT—(Double.)		FLY-WHEEL.		Diameter Steam Pipe, Inches.	Diameter Exhaust Pipe, Inches.
Diameter, Inches.	Stroke, Inches.			Diameter, Feet.	Face, Inches.	Weight, Pounds.	Width, Inches.	Velocity Ft. per Min.	Diameter, Feet.	Weight, Pounds.		
6	12	200	3	5	8	500	7	3140	6	600	$1\frac{1}{2}$	2
7	12	200	$3\frac{1}{2}$	5	8	700	7	3140	6	800	2	3
8	12	200	4	5	9	800	8	3140	5	900	2	3
9	16	175	$4\frac{1}{2}$	6	10	1000	9	3297	7	1200	$2\frac{1}{2}$	$3\frac{1}{2}$
10	16	175	5	6	11	1500	10	3297	8	1700	$2\frac{1}{2}$	$3\frac{1}{2}$
12	20	150	6	8	13	2500	12	3770	9	3000	3	4
14	20	150	7	8	15	3000	14	3770	10	3600	$3\frac{1}{2}$	5
15	24	140	$7\frac{1}{2}$	9	18	4000	16	3958	12	4800	4	5
16	24	140	8	9	20	5000	18	3958	12	5800	4	6

The belt powers in the above table are for three-eighth cut off, with 90 pounds steam, or for the ordinary power ratings when fitted with the common D valve, cutting off at about five-eighths, and throttling with the governor.

THE CUMMER ENGINE CO.,

CLEVELAND, O.

Class D. Slide-Valve Engines with fixed Cut-off.

CYLINDER.		Rev'lut'ns per Minute.	Piston Speed in Ft. per Min.	Horse Power constant for 1 lb. M. E. P.	HORSE POWER.		
Diameter Inches.	Stroke Inches.				30 M. E. P.	35 M. E. P.	40 M. E. P.
6	12	200	400	.3427	10.3	12.0	13.7
7	12	200	400	.4664	14.0	16.3	18.7
8	12	200	400	.6092	18.3	21.3	24.4
9	16	175	467	.9002	27.0	31.5	36.0
10	16	175	467	1.1115	33.4	38.9	44.5
12	20	150	501	1.7169	51.5	60.1	68.7
14	20	150	501	2.3369	70.1	81.8	93.5
15	24	140	560	2.9478	88.4	103.2	117.9
16	24	140	560	3.4119	102.4	119.4	136.5

The cut-off is fixed at ¼ stroke in all cases here given.

The regular steam pressures to be carried for the mean effective pressures as given in the above table are as follows:

30 lbs. M. E. P. = 81 lbs. pressure per sq. inch in the boiler.
35 " " = 94 " " " "
40 " " = 106 " " " "

For engines with a fixed cut-off only ⅘ of the boiler pressure is available, and for this reason the boiler pressures in the above table are higher than for the corresponding M. E. P. for automatic cut-off engines.

THE CUMMER ENGINE CO.,

CLEVELAND, O.

Class D. Slide-Valve Engines with fixed Cut-off.

CYLINDER.		Rev'lut'ns per Minute.	Piston Speed in Ft. per Min.	Horse Power constant for 1 lb. M. E. P.	HORSE POWER.		
Diameter Inches.	Stroke Inches.				40 M. E. P.	45 M. E. P.	50 M. E. P.
6	12	200	400	.3427	13.7	15.4	17.1
7	12	200	400	.4664	18.7	21.0	23.3
8	12	200	400	.6092	24.4	27.4	30.5
9	16	175	467	.9002	36.0	40.5	45.0
10	16	175	467	1.1115	44.5	50.0	55.6
12	20	150	501	1.7169	68.7	77.3	85.9
14	20	150	501	2.3369	93.5	105.2	116.9
15	24	140	560	2.9478	117.9	132.7	147.4
16	24	140	560	3.4119	136.5	153.5	170.6

The cut-off is fixed at ⅜ stroke in all cases here given.

The steam pressures to be carried for the mean effective pressures as given in the above table are as follows:

40 lbs. M. E. P.=75 lbs. pressure per sq. inch in the boiler.

45 " " =88 " " " "

50 " " =94 " " " "

This is on the basis of $\frac{4}{5}$ of the boiler pressure being available in the cylinder when using a fixed cut-off.

THE CUMMER ENGINE COMPANY,

CLEVELAND, O.

Class E. Self-Contained Side-Valve Engines.

GENERAL DIMENSIONS.	HORSE POWER.					
	6	8	10	15	20	25
Diameter of Cylinder (in.)	5	6	7	8	9	10
Length of Stroke (in.)	8	8	10	10	12	12
Revolutions per Minute	220	220	200	200	180	180
Flywheel, Diameter (in.)	36	40	44	48	52	56
" Face (in.)	6	6	7	8	9	10
Pulley, Diameter (in.)	16	16	18	20	22	24
" Face (in.)	8	9	8	8	9	10
Steam Pipe, Diameter (in.)	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	$2\frac{1}{2}$
Exhaust Pipe " "	2	2	2	$2\frac{1}{2}$	3	3

Each Engine will be furnished with flywheel and pulley of the dimensions given above, governor and belt, stop valve, drain cocks, anchor plates, foundation bolts, oil cups and wrench.

TABLE OF MEAN-EFFECTIVE AND TERMINAL PRESSURES

—— OF THE ——

CUMMER CONDENSING & NON-CONDENSING ENGINES.

PRESSURES. Above Atmos.	PRESSURES. Above Vac'um	Point of Cut-Off.	Rate of Expan.	Mean pressure througho't Stroke.	BACK PRESSURE. Non-con'g	BACK PRESSURE. Conde'g.	Mean Effective Press. Non-con'g	Mean Effective Press. Condens'g	Terminal pressure above Vacuum.
50	65	$\frac{1}{5}$	5	33.77	16	4	17.77	29.77	12.94
		$\frac{1}{4}$	4	38.56	16	4	22.56	34.56	16.18
		$\frac{3}{8}$	2.66	48.14	16	4	32.14	44.14	24.26
		$\frac{1}{2}$	2	54.74	16	4	38.74	50.74	32.35
55	70	$\frac{1}{5}$	5	36.38	16	4	20.38	32.38	13.94
		$\frac{1}{4}$	4	41.54	16	4	25.54	37.54	17.43
		$\frac{3}{8}$	2.66	51.86	16	4	35.86	47.86	26.14
		$\frac{1}{2}$	2	58.97	16	4	42.97	54.97	34.85
60	75	$\frac{1}{5}$	5	38.99	16	4	22.99	34.99	14.94
		$\frac{1}{4}$	4	44.52	16	4	28.52	40.52	18.68
		$\frac{3}{8}$	2.66	55.58	16	4	39.58	51.58	28.01
		$\frac{1}{2}$	2	63.20	16	4	47.20	59.20	37.35
65	80	$\frac{1}{5}$	5	41.60	16	4	25.60	37.60	15.94
		$\frac{1}{4}$	4	47.50	16	4	31.50	43.50	19.93
		$\frac{3}{8}$	2.66	59.30	16	4	43.30	55.30	29.89
		$\frac{1}{2}$	2	67.43	16	4	51.43	63.43	39.85
70	85	$\frac{1}{5}$	5	44.21	16	4	28.21	40.21	16.94
		$\frac{1}{4}$	4	50.48	16	4	34.48	46.48	21.18
		$\frac{3}{8}$	2.66	63.02	16	4	47.02	59.02	31.76
		$\frac{1}{2}$	2	71.66	16	4	55.66	67.66	42.35
75	90	$\frac{1}{5}$	5	46.82	16	4	30.82	42.82	17.94
		$\frac{1}{4}$	4	53.46	16	4	37.46	49.46	22.43
		$\frac{3}{8}$	2.66	66.74	16	4	50.74	62.74	33.64
		$\frac{1}{2}$	2	75.89	16	4	59.89	71.89	44.85
80	95	$\frac{1}{5}$	5	49.43	16	4	33.43	45.43	18.94
		$\frac{1}{4}$	4	56.44	16	4	40.44	52.44	23.68
		$\frac{3}{8}$	2.66	70.46	16	4	54.46	66.46	35.51
		$\frac{1}{2}$	2	80.12	16	4	64.12	76.12	47.35
85	100	$\frac{1}{5}$	5	52.04	16	4	36.04	48.04	19.94
		$\frac{1}{4}$	4	59.42	16	4	43.42	55.42	24.93
		$\frac{3}{8}$	2.66	74.18	16	4	58.18	70.18	37.39
		$\frac{1}{2}$	2	84.35	16	4	68.35	80.35	49.85
90	105	$\frac{1}{5}$	5	54.65	16	4	38.65	50.65	20.94
		$\frac{1}{4}$	4	62.40	16	4	46.40	58.40	26.18
		$\frac{3}{8}$	2.66	77.90	16	4	61.90	73.90	39.26
		$\frac{1}{2}$	2	88.58	16	4	72.58	84.58	52.35
95	110	$\frac{1}{5}$	5	57.26	16	4	41.26	53.26	21.94
		$\frac{1}{4}$	4	65.38	16	4	49.38	61.38	27.43
		$\frac{3}{8}$	2.66	81.62	16	4	65.62	77.62	41.14
		$\frac{1}{2}$	2	92.81	16	4	76.81	88.81	54.85
100	115	$\frac{1}{5}$	5	59.87	16	4	43.87	55.87	22.94
		$\frac{1}{4}$	4	68.36	16	4	52.36	64.36	28.68
		$\frac{3}{8}$	2.66	85.34	16	4	69.34	81.34	43.01
		$\frac{1}{2}$	2	97.04	16	4	81.04	93.04	57.35

In order to get rid of an inconvenient fraction in the second column the pressures above the atmospheric are given in round numbers at 15 pounds instead of 14.7 pounds. The calculations are all made with the latter and not the former figures.

EASE WITH WHICH ENGINES MAY BE SET UP.

Although in many cases we send men to attend to the erecting and starting of our engines, yet this is by no means necessary and frequently the purchasers are able to do all such work themselves. We send working drawings for the foundations of engines, for the boiler settings and the general arrangement of the engine and boiler rooms. There is often in the employ of the purchaser a millwright, or other expert in setting up machinery, who is competent to superintend the construction of foundations as well as to level and square up the engine. After this but little remains to be done. Our engines are all, to a great extent, self-contained and it is thus an easy matter to place them upon the foundations and line them up. When erected in the shop, all the parts of the engines are plainly marked, so that when taken apart no difficulty need be experienced in putting them together again ; but the engines are shipped in such condition that very little has to be done in this way when they are being erected.

Before leaving the shop the valves are properly set and the valve-stems and eccentric rods adjusted to proper length ; the valves, valve-stems, &c., remain in position when the engines are shipped while the eccentric-rods are removed, but before taking them off their lengths are carefully trammed, and centre punch marks made, so that by means of a tram, which is sent with the engine, it is a very simple matter to screw in the rods so as to have exactly the same lengths as before, and the valves themselves are then properly set ready for starting up the engine. This tram is to be retained and used from time to time in order to test the lengths of the rods and see that everything is in proper adjustment.

ALWAYS KEEP US INFORMED HOW OUR ENGINES ARE WORKING.

The demand of the intelligent manufacturer is that his engine shall be simple, durable, economical, closely governed, not liable to get out of order and easily managed. In the foregoing pages of this catalogue, we have aimed to give so plain a description of our engine that all who read carefully may be able to understand its construction and to decide for themselves with reference to its merits. We have brought to bear upon the design of our engine, the results of long study and experience in this field, and, recognizing the fact that correctness of principle and excellence of workmanship are equally important, we have also exercised the same intelligent care in the design and selection of tools and machinery specially adapted for its proper construction. The merits of our engine, and our careful attention to its manufacture, have already gained for us an excellent reputation which we are naturally desirous

shall be maintained and increased. Our interest in our engines does not cease when they leave our hands and payment has been made, and even though years should elapse, we feel that we cannot afford to have any of our engines running unless they are working creditably and giving satisfaction to their owners. We appreciate the fact that an engine may be correct in every detail of its design and construction and yet occasionally fail to give its owner the results that he may naturally expect and may also fail to obtain for itself and us such credit as is fully deserved. It sometimes happens, at the outset, that foundations are not well built or that they settle after the engine has been placed in position; the valves may not be properly adjusted when starting up or they may afterwards get out of adjustment. Occasionally engines are placed in the hands of incompetent men and there are various ways in which trouble can occur. The cause of any difficulty, even with all our care, might sometimes be with ourselves through the oversight of our employees. But let it be where it may, we should like always to be informed, by the owner of any of our engines, whenever he is not entirely satisfied with the working of his machine; and it will, in the majority of cases, occur that a few questions from us will lead to a discovery of the cause of difficulty and enable us by letter to suggest the proper remedy.

If after an engine has been in use some time the economy appears less satisfactory than it was at first the trouble perhaps is with the boiler. There may be several causes for this of which the one most likely to occur is that, through neglect or want of proper care and attention, scale has been allowed to form in the boiler to an injurious extent. It might also happen that the engine was working under a lighter load, or a heavier load, than was originally intended. In either event, the cut-off would not be at an economical point, and, where a much heavier load than the proper one was being used, the boiler would no longer be large enough for its work. In order to make enough steam for its increased duty the fires have to be forced, this makes it more difficult to maintain regular pressure and to obtain dry steam, while the larger amount of water which must be evaporated causes an excessive deposit of scale, and not only this, but there is not the same opportunity for thorough and complete combustion, which is given where a boiler is sufficiently large to do its regular work without forcing.

Sometimes we furnish only the engine while the boilers are of some other make and, perhaps, these boilers are old ones which had been used for the engine displaced by ours. Where lack of economy is complained of, in this case, it will frequently be found to be in consequence of a boiler whose size is too small for its work, or, because of some incorrect proportions in the design of the boiler. There may be defective draft;

there may be too much grate surface, or too little. The amount of heating surface may be greater or less than is required ; or, it may be so badly arranged that proper circulation of the water is prevented and that no facilities are afforded for examinations and removal of scale. Any of these causes, and others which could be mentioned, will interfere with economy ; but we will not go into this subject here any further than to add that, for extreme economy, the engine in the first place should be of proper size for the average load and then, from the beginning to the end of a power outfit, there should be an intelligent harmony in the proportions of each part, one to the other, and all to be supplemented by intelligent management.

WEIGHTS OF WROUGHT IRON SHAFTING.

Diam. of Shafting.	Weight in Lbs. of 1 Ft.		Diam. of Shafting.	Weight in Lbs. of 1 Ft.	
	Finish'd Sizes	Rough Sizes.		Finish'd Sizes	Rough Sizes.
$1\frac{1}{2}$	5.41	5.89	$3\frac{1}{4}$	26.60	27.65
$1\frac{3}{4}$	7.46	8.02	$3\frac{1}{2}$	30.94	32.07
2	9.83	10.47	4	40.59	41.89
$2\frac{1}{4}$	12.53	13.25	$4\frac{1}{2}$	53.01	56.00
$2\frac{1}{2}$	15.55	16.36	5	65.45	68.76
$2\frac{3}{4}$	18.91	19.80	$5\frac{1}{2}$	79.19	82.83
3	22.59	23.56	6	94.25	98.22

NOTE.—Up to 4″ the finished sizes are $\frac{1}{16}$″ less than the nominal diameters in the rough. Above 4″ the finished diameters are on size, but the weights in the rough are based upon $\frac{1}{8}$″ larger diameter.

BOILERS.

We are prepared to furnish with our engines any kind of boiler that may be desired; the particular form of boiler, which is best suited to the requirements of any given case, will vary according to circumstances. Our various tables give dimensions and horse-powers of several forms of boilers, any one of which will yield excellent results, and they are such as we recommend to be used with our engines. Our boilers are all made of first-class material; there is an ample factor of safety allowed; the longitudinal seams are double riveted, and the flat surfaces are thoroughly braced. All boilers are carefully inspected and tested to a pressure of 150 lbs. per square inch before they are sent out. We prefer to use steam of 90 or 100 lbs. pressure, and the boilers are constructed to withstand this amount. Sometimes persons are fearful lest these pressures may be dangerous, and wish to carry only 50 or 60 lbs., but 90 or 100 lbs. is not, by any means, a high pressure, and such boilers, properly constructed, will really be more secure than very many which are working at a pressure of 50 or 60 lbs. It is merely a question of strength, and is easily provided for, while the extra care used in the manufacture renders these boilers so safe that they may be used with full confidence. There are mainly two things to be attended to in the design and selection of a boiler; the first is to have a boiler constructed with a view to economy in fuel, and the other is to have the boiler adapted to whatever kind of feed water is to be used. In most cases economy is desirable, and this depends principally upon securing complete combustion, and then having a sufficient amount of properly arranged heating surface, to take up the heat and convey it to the water. Where scale is not likely to be formed upon the tubes, and other heating surfaces, one of our tubular boilers, with 3, 3½ or 4-inch tubes will be found a very compact and economical steam generator. Where impure, muddy water or hard water is employed, it may be necessary to select a boiler of a different kind, because the scale and deposits of foreign matter upon the heating surfaces will reduce the efficiency so much, and is moreover so dangerous, because of the risk of blistering or even burning out the tubes and plates, that for such cases a boiler must be adapted in its construction so as to admit of easy access to all parts of the interior to remove the incrustations. One of our tubular boilers, with a central space, such as shown in Fig. 37, is recommended where only a moderate amount of scale is liable to be formed, these boilers are better adapted for ready inspection and removal of scale, than the ordinary tubular boiler, and there is also a much better circulation of the water. Boilers still better adapted for working with bad water are the six-inch tubular, and those of the five-flue variety. Five-flue boilers are made with 7, 8, 9 or 10-inch tubes, according to the diameter of the shell. The tubes are either made of plates, with

riveted seams or what is still better, they are seamless lap-welded tubes. Owing to the danger of collapse, which increases with the diameter, and also the length of the flue, it is not advisable to have them either of too large a diameter or too great a length. But the flues, such as given in our table, are not of large diameter, and with their ample thickness and moderate length, there is no danger to be apprehended from this cause. Our six-inch tubular boilers, may be considered intermediate between the ordinary tubular boiler and the five-flue boilers; the tubes, in consequence of being riveted to flanged openings in the boiler heads, cannot be so closely spaced as is done on the three or four-inch tubular boilers. This larger space renders this kind of boiler one which is well suited to cases where bad water causes incrustations and makes it desirable to have easy access to the tubes for cleaning, and to allow proper circulation, even if scale has formed to some extent upon the tubes. When lime extracting heaters or other means of purifying the feed water, are used, or recourse is had to various boiler compounds, in order to prevent or remove scale, then there is less objection to using a tubular boiler with 3, 3½ or 4-inch tubes, and thus there may be secured better economy in fuel than is possible with the other forms of boilers having less heating surface. In order to have access to the interior, man holes are always provided in each head; whenever a boiler is large enough to admit of it, there should be a man-hole above, and one below the tubes or flues; where this is not possible, the man-hole is placed above, and a hand-hole below; this, in general, will give ample facilities for inspection, cleaning and repairs.

MATERIALS USED IN BOILERS.

The materials used in boiler making are cast iron, wrought iron and steel. Cast iron is used for fronts, grate bars, brackets, columns and stands for supporting the boiler, for nozzles, back plates for arches, at the rear of boiler, and for many other purposes; but, except in some forms of sectional boilers, cast iron is not used for the boiler itself. Man-holes should always be surrounded with a heavy wrought iron ring, securely riveted to the plate in which the hole is made; and this is demanded in order to compensate for the weakening effect of cutting such large holes in the shell or head of a boiler. The safety of a steam boiler, involving as it does that of human life and valuable property, nothing but thoroughly reliable materials should be allowed to enter into its construction, and one of the very first considerations in the design of a boiler should be to have it perfectly safe. Wrought iron, in the form of plates of various thicknesses, and of lap welded or drawn tubes is the material most commonly employed; mild steel is being gradually introduced, and there is hardly a question, that at no distant day, it will be very largely used. The grades of wrought iron plate, suitable for boiler making and

manufactured expressly for the purpose, are known by the following brands: C. H. No. 1 Shell Iron; C. H. No. 1 Flange Iron; and C. H. No. 1 Fire-box Iron. These irons are all made from very pure ores, and in all the operations of smelting, refining, and heating, regard is had to the purity of the fuel which is used, since this largely affects the quality of the iron. Great care is exercised in hammering and rolling the slabs, and the piles, which are made into plates, in order to ensure thorough working and remove all slag and cinder, so that the iron produced may be perfectly welded and free from seams and laminations, and have a high tensile strength, combined with ductility. Iron of this kind is fitted to resist the strains to which a boiler is subjected, to withstand the effect of heat, and to undergo all the operations of punching, shearing, etc., without being injured in quality. It is not possible for an inferior iron to stand the tests for tensile strength and ductility, and hence these qualities, which may be easily determined by actual test with a testing machine, may be accepted as deciding whether boiler plate is good or bad. Many makers stamp their plates with their names, and the tensile strength of the iron. Reputable makers will not stamp plates unless they are sure the quality is as represented; in the absence of any brand it is a fair inference that the quality is not good enough for the maker to be willing to own his production.

C. H. No. 1 Shell Iron is used for the shells of boilers; it should never be used for the heads, because its quality is not good enough to admit of flanging. The tensile strength ranges from 45,000 lbs. to 60,000 lbs. per square inch. Good boiler plate ought not to have a less tensile strength than 50,000 lbs. per square inch; this strength, and even more, is yielded by all the well-known brands of this grade of iron.

C. H. No. 1 Flange Iron is made in a similar manner as C. H. No. 1 Shell, but the quality of iron used is better, and the working more thorough; the result is a softer, purer iron, less fibrous in structure, and of greater tensile strength and toughness; it is also nearly as strong across the grain, as in the direction of the grain, which is not the case with C. H. No. 1 Shell Iron. Flange iron should never have a less tensile strength than 50,000 lbs.; it reaches 60,000 lbs. or 65,000 lbs., but seldom exceeds this latter figure. Too great a tensile strength must be purchased at the expense of other desirable qualities, as ductility and toughness. This quality of iron will stand repeated heating and cooling without becoming brittle, and is well adapted for being flanged, as is required for boiler heads, and should always be used for that purpose.

C. H. No. 1 Fire-box Iron is similar in quality to that just described, except that it is harder and better able to resist the high temperature of the fire-box, and should be used in such situations; the tensile strength is high, and the metal admits of flanging, which is often desirable in forming the joints of a fire-box.

STEEL FOR BOILER PLATES.

A better material than wrought iron plate, and one which will, no doubt, have a very extended use in the near future, is boiler plate made of mild steel. This metal may be produced by either the Crucible, Bessemer or Siemens-Martin process, of which the preference is to be given to the latter. Although termed steel, the carbon in this metal is not sufficient to cause it to harden, and ingot iron, a name which has been proposed, would be a more correct word. It is really a very pure wrought iron, with only a very small proportion of carbon; the quantity permitted in plates suitable for boilers is from .1 to .15 of one per cent. The addition of a larger amount will give more tensile strength, but it decreases the ductility and toughness, and beyond a certain amount will cause the metal to harden. Steel adapted for boilers should have a tensile strength of from 60,000 to 65,000 pounds per square inch, and should stretch some 20 to 30 per cent. before breaking. Steel has an advantage over wrought iron in being of a homogeneous structure; it is, therefore, nearly equally strong in every direction, and being almost entirely free from any laminations, it is well fitted to withstand the flame and heat of the furnace or fire-box. Wrought iron plates are made by piling together and rolling slabs of metal which were previously hammered or rolled, and the quality of the plate produced depends not only upon the original purity of the iron, but upon the thoroughness with which slag or cinder is eliminated, and the whole mass welded into a solid plate free from laminations and blisters, which are the result of imperfect welding. Wrought iron has always more or less fibre, and is not so strong across the grain, as in the direction of the grain, nor do any but the very best qualities admit of flanging equally well in any direction. But steel having been thoroughly melted and poured when in a very fluid state, and the resulting ingot hammered and rolled, or simply rolled, into boiler plate, the metal has a much more uniform texture, and is in every way better suited to its purpose than wrought iron plates. Steel has a tensile strength, which is determined largely by the per centage of carbon it contains; but this is not to be accepted without qualification. The strength and many physical properties of the metal depend largely upon the arrangement of its particles, and this is determined by the treatment it has received in the various operations to which it has been subjected; test bars from steel of exactly the same chemical composition will respond differently to various tests, accordingly as they have been treated differently. It is found, also, that steel plates cannot be worked in the boiler shop according to the same methods as are employed with wrought iron. The operations of punching, shearing and flanging are more injurious to the plate, but it is found that the original good qualities may be

restored by annealing, and this should always be done. The point to be noted particularly is that the ordinary processes, as used for boilers of wrought iron plate, must be modified when boilers are to be made of steel, and that when properly constructed, according to the principles which have been found effective in working steel, the element of uncertainty in the quality of this material is eliminated; and, the boilers may be used with fully as much confidence as if they were made of wrought iron, while there is secured all the superior advantages which are possessed by steel plates.

CIRCULATION OF WATER.

The transfer of heat in steam boilers is mainly accomplished by what is known as convection. The simple and familiar experiment of placing some particles of saw dust or bran in a glass vessel, partially filled with water, and applying heat beneath the vessel, clearly illustrates what takes place on a larger scale in a steam boiler. The small particles in the vessel, as the heat increases, will soon set up an upward and downward movement; this is caused by those portions immediately in contact with the source of heat becoming more highly heated than those more remote; the hotter water rises because its density is less than the colder water which displaces it. The cold water, in turn, becomes heated and is displaced by another and colder portion; thus a continual movement of the water takes place, until the temperature rises sufficiently for ebullition to begin, and steam is then given off, which goes on until all the water has been evaporated. Now, it will be evident that if anything had been allowed to hinder the free movement of these currents we speak of, that the transfer of heat would have been less rapid and thorough; and, since in a steam boiler the same action takes place, especial care must be exercised that these movements of the water, by which is meant the circulation, are not impeded to an injurious extent. The plain cylinder boiler, having nothing but a clear space on the inside, admits of a better circulation than a tubular or flue boiler, because, just as soon as tubes or flues are introduced, obstructions are offered to the movement of the water and the bubbles of steam which form around the heating surfaces. It is from this cause that knowledge and good judgment must be exercised in the spacing of tubes, and there can be no greater mistake than to suppose, without qualification, that a large extent of heating surface means a correspondingly large power for the boiler. If the tubes are crowded together too closely it will interfere so much with the circulation of the water and the generation of the steam, that a positive improvement may be effected by taking out some of the tubes, so as to give more space. Another matter which must be looked to is that, when in use and deposits of scale accumulate on the tubes, the space between them is

lessened, and enough space to compensate for this must be allowed at the outset. Our boiler tables give the diameter and number of tubes properly spaced for each boiler, so as to permit thorough circulation of the water, and free generation of the steam. The tubes are arranged in vertical rows, so that there is a clear, straight space between each row, as shown in Fig. 36. Tubes should not be arranged zig zag or so that the tubes in one horizontal row are placed over the spaces in the adjacent rows. Where water is liable to form scale, and thus impede circulation when tubes are spaced in the ordinary way, it is better to adopt the method shown in Fig. 37, and designated in our tables as being arranged with a central space. Here there is a much better opportunity for a free movement of the water and bubbles of steam, as they are

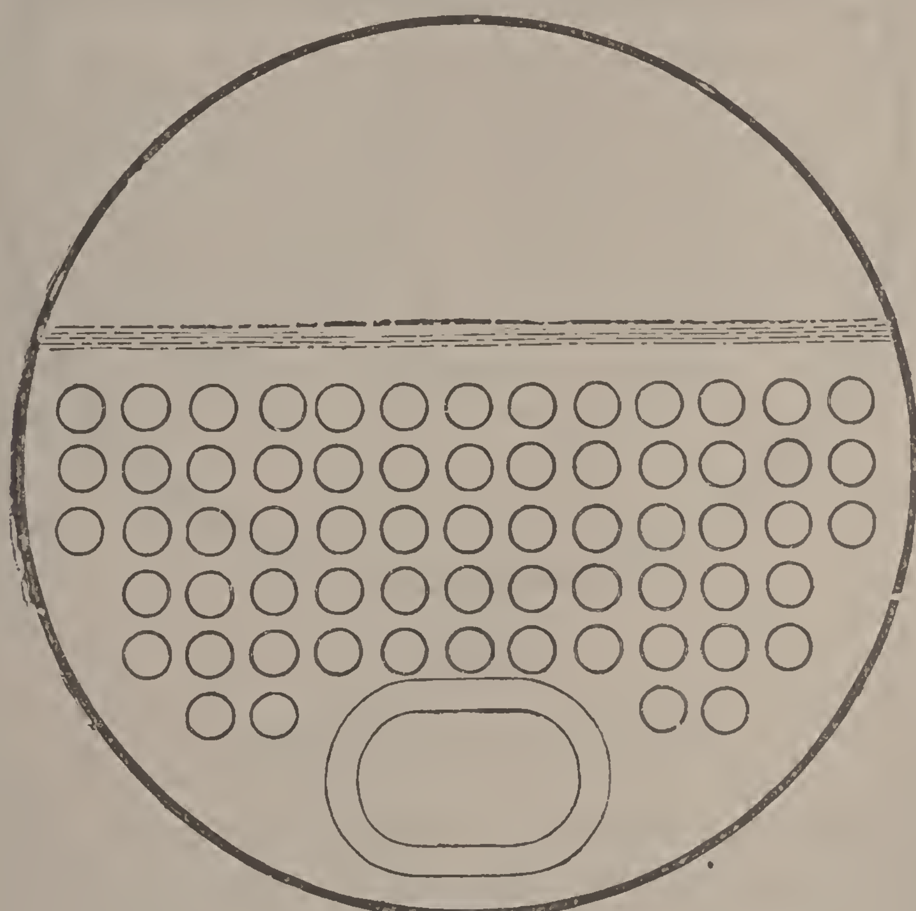


Fig. 36.

disengaged from around the heating surfaces; boilers of this kind will be found most efficient where there is liability for the tubes to become covered with scale, and they should be used in such cases. More or less distance is allowed between the vertical rows on either side of the central space, according to the diameter of the shell; and, since the tubes are otherwise spaced the same as in the ordinary boiler, there is thus a less number of tubes for a boiler of given diameter, as will be seen by comparing the tables for each method of arrangement; but the increased efficiency consequent on the more thorough circulation, and which is especially valuable where water is impure, will more than compensate for the reduction in heating surface caused by removing a

row of tubes from the centre. The general movement of the water in a boiler of this kind will be up the central space and down on each side, or the reverse, accordingly as either the bottom or the sides of the shell are more highly heated ; this movement goes on continually as long as heat is applied, and renders this form of boiler an excellent one in many respects.

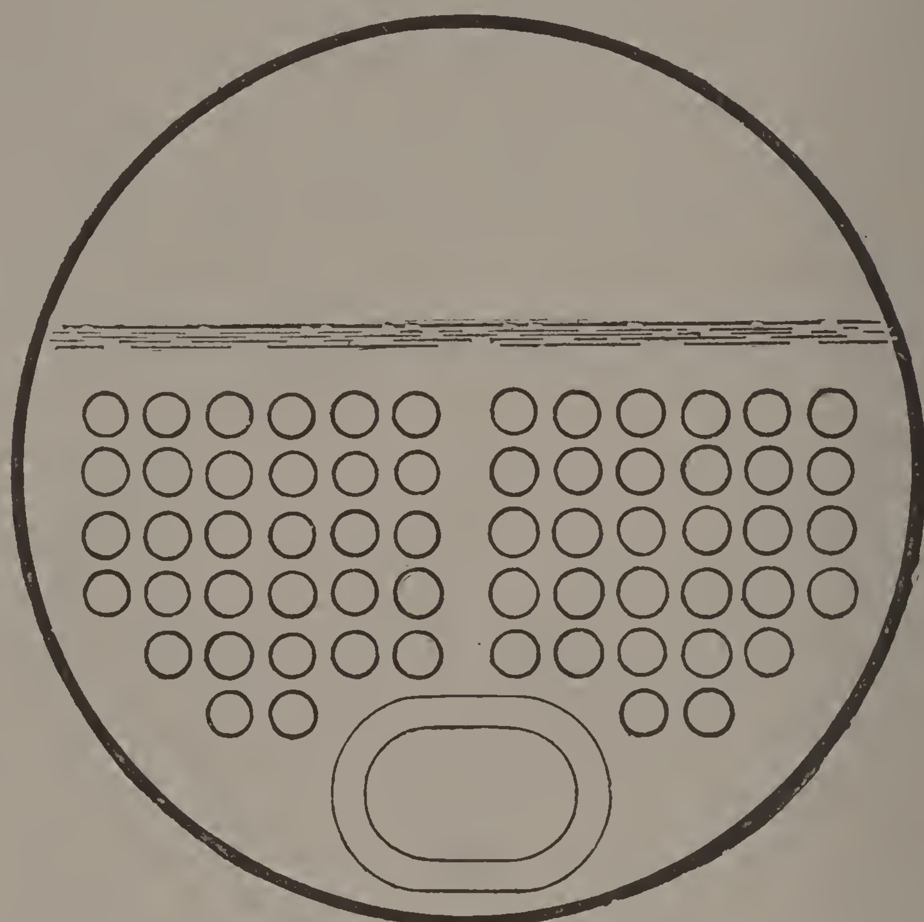


Fig. 37.

HEATING SURFACES.

Heating surfaces in boilers are all those surfaces which come in contact with the radiant heat and hot gases from combustion on the one side, and with the water on the other. For internally fired boilers, the sides and crown sheet or top of the fire-box, together with the combined surface of all tubes or flues is the heating surface. For boilers in which the furnace is external, the total area of all the tubes or flues, and a certain proportion of the shell, which we have taken at $\frac{2}{3}$ the area, will constitute the whole heating surface. It will be apparent, upon slight consideration, that all surfaces reckoned as heating surfaces, are not equal in value, merely because their extent in square feet is the same. Heat is transmitted to the water in proportion to the difference in temperature between that due to the source of heat on one side of the plate, and of the water on the other side. Those parts of the boiler, therefore, which are exposed to the high temperature of the fire-box, in an internally fired boiler, have a much higher efficiency than an equal number of

square feet in the tubes of an ordinary boiler; the amount of heat transmitted per square foot is very much greater, and it would not be at all fair to make simple extent of heating surfaces the ground for comparison between two such boilers. So, also, with flue boilers, the heating surface of the flues is more effective than an equal amount in the tubes of a tubular boiler.

It will be manifiest, then, that no intelligent idea can be formed of the horse-power of a boiler by simply knowing the extent of its heating surfaces; it is necessary, also, to take into account the effectiveness of the heating surface, which, as we have already seen, varies greatly in the different types of boilers. Another point which involves a common mistake, but to which we will merely allude in this connection, is that, with a given boiler, merely increasing the amount of heating surface, by putting in more tubes. does not necessarily increase the horse-power or capacity to generate steam; indeed, the very reverse may take place, because the tubes are then so crowded together that proper circulation of the water, upon which the generation of steam depends, is not permitted, and therefore the boiler, even with all its heating surface, is not able to make steam as rapidly as it would do if the tubes were properly spaced, so as to allow the ascending and descending currents of water, and the bubbles of steam which form around the heating surfaces, to flow freely without being impeded in their progress by impinging against the tubes. But, while simple extent of heating surface does not, in itself, convey a just notion of the capacity of a boiler as a steam generator, it is yet necessary to have some standard of reference, in order to reckon the horse-power, and we have taken the ordinary tubular boiler as a representative form. According to the effectiveness of the surface, other kinds of boilers will require more or less than the number of square feet allowed per horse-power with tubular boilers, and this must always be so understood when any comparison of boilers is to be made. But besides a knowledge of the effectiveness of the heating surface, it is necessary, when judging whether the allowance is sufficiently liberal, to know what kind of engine is to be used with the boiler; and when we say that so many square feet of heating surface are allowed for each horse-power, it must be understood that this quantity is varied according to the class of engine for which the boiler is proportioned; and, that what will be ample for one kind of engine will, perhaps, be only one-half or one-third what is required by another. Thus, for a moderately large compound condensing engine, steam-jacketed, provided with an ordinary and an auxiliary heater, and all precautions against waste of heat, five square feet of heating surface, with a good tubular boiler, will be found a sufficient allowance. For an automatic condensing engine, not even provided with more than the one heater, seven square feet of heating surface per

horse-power will be ample. For an automatic non-condensing engine, 10 square feet of heating surface is a good allowance. Slide valve engines, with fixed cut-off, such as those of Class D, do not need more than 12 square feet of heating surface, while plain slide valve engines, especially if they be of small size will require 15 square feet, and in some cases even more. In general, engines of large size will need less heating surface per horse-power than small engines; and, in proportion as the heating surface is reduced from this cause, and also from having a more efficient engine, so is the amount of coal lessened per horse-power per hour. Thus with an engine of the highest grade, where 5 or 6 square feet of heating surface is allowed to the horse-power, the consumption of coal is two pounds or less per horse-power per hour; while, where 15 or more square feet are allowed, the consumption of coal is 5, 6 or more pounds per horse-power per hour. The figures just given are for engines of medium size, and these amounts of heating surface, required by different classes of engines, are not derived from merely theoretical considerations, but are based upon the results of our own actual and successful practice.

HORSE-POWER OF BOILERS.

The term horse-power, as commonly used to describe the size of a boiler, and its capacity to make steam is, it must be admitted, somewhat vague and unsatisfactory, as well as plainly incorrect. A horse-power, strictly speaking, means a rate of mechanical work, and has no other meaning. But it is necessary, in proportioning boilers to engines of certain size and power, to have some convenient way of designating them, and since the amount of water evaporated by well proportioned boilers of a given type is according to the extent in square feet of their heating surfaces it has become customary to allow so many square feet of heating surface to be sufficient for evaporating the quantity of water required for each horse-power. Thus, if 10 square feet be allowed for each horse-power, the total heating surface of the shell and tubes divided by 10 gives the horse-power of the boiler; so also, if seven square feet or 12 square feet be considered the proper amount for a horse-power, the total heating surface is to be divided by seven or by twelve, in order to obtain the horse-power. In our boiler tables it will be noticed that there are three columns under the heading horse-power, in which the horse-powers of the boilers are based upon 7, 10 or 12 square feet of heating surface respectively. The quantity of water required per horse-power per hour, varies according to the class of engine; and, therefore, the amount of heating surface, which is necessary to evaporate a given quantity of water, will be less for a high grade engine than it is for an ordinary engine. This difference must be borne in mind when selecting from the table, a boiler of proper size for an engine of a certain kind and horse-power, because, the horse-power of the boiler must be based upon 7, 10 or 12 square feet of heating surface, accordingly as the one or the other amount may be demanded by the particular class of engine for which a suitable boiler is desired.

Three-Inch Tubular Boilers.

TUBES ARRANGED WITHOUT A CENTRAL SPACE.

BOILER.			SQUARE FEET OF HEATING SURFACE.			HORSE POWER. BASED UPON		
Diam. Inches.	Length. Feet.	No. of Tubes.	Shell.	Tubes.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
36	10	19	62.8	149.2	212.0	30.3	21.2	17.7
	12	19	75.4	179.0	254.4	36.3	25.4	21.2
	14	19	88.0	208.9	296.9	42.4	29.7	24.7
	16	19	100.5	238.7	339.2	48.5	33.9	28.3
38	10	26	66.3	204.2	270.5	38.6	27.1	22.5
	12	26	79.6	245.0	324.6	46.4	32.5	27.0
	14	26	92.8	285.9	378.7	54.1	37.9	31.6
	16	26	106.1	326.7	432.8	61.8	43.3	36.1
40	10	32	69.8	251.3	321.1	45.9	32.1	26.8
	12	32	83.8	301.6	385.4	55.1	38.5	32.1
	14	32	97.7	351.8	449.5	64.2	45.0	37.5
	16	32	111.7	402.1	513.8	73.4	51.4	42.8
42	10	32	73.3	251.3	324.6	46.4	32.5	27.0
	12	32	88.0	301.6	389.6	55.7	39.0	32.5
	14	32	102.6	351.8	454.4	64.9	45.4	37.9
	16	32	117.3	402.1	519.4	74.2	51.9	43.3
44	10	38	76.8	298.5	375.3	53.6	37.5	31.3
	12	38	92.2	358.2	450.4	64.3	45.0	37.5
	14	38	107.5	417.9	525.4	75.1	52.5	43.8
	16	38	122.9	477.6	600.5	85.8	60.1	50.0
46	10	42	80.3	329.9	410.2	58.6	41.0	34.2
	12	42	96.3	395.9	492.2	70.3	49.2	41.0
	14	42	112.4	461.9	574.3	82.0	57.4	47.9
	16	42	128.5	527.8	656.3	93.8	65.6	54.7
48	10	46	83.8	361.3	445.1	63.6	44.5	37.1
	12	46	100.5	433.6	534.1	76.3	53.4	44.5
	14	46	117.3	505.8	623.1	89.0	62.3	51.9
	16	46	134.0	578.1	712.1	101.7	71.2	59.3
50	10	55	87.3	432.0	519.3	74.2	51.9	43.3
	12	55	104.7	518.4	623.1	89.0	62.3	51.9
	14	55	122.2	604.8	727.0	103.9	72.7	60.6
	16	55	139.6	691.2	830.8	118.7	83.1	69.2
52	10	46	90.8	361.3	452.1	64.6	45.2	37.7
	12	46	108.9	433.6	542.5	77.5	54.3	45.2
	14	46	127.1	505.8	632.9	90.4	63.3	52.7
	16	46	145.2	578.1	723.3	103.3	72.3	60.3
54	10	52	94.2	408.4	502.6	71.8	50.3	41.9
	12	52	113.1	490.1	603.2	86.2	60.3	50.3
	14	52	132.0	571.8	703.8	100.5	70.4	58.6
	16	52	150.8	653.4	804.2	114.9	80.4	67.0

THREE-INCH TUBULAR BOILERS.—Continued.

TUBES ARRANGED WITHOUT A CENTRAL SPACE.

BOILER.			SQUARE FEET OF HEATING SURFACE.			HORSE POWER. BASED UPON		
Diam. Inches.	Length. Feet.	No. of Tubes.	Shell.	Tubes.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
58	10	62	101.2	486.9	588.1	84.0	58.8	49.0
	12	62	121.5	584.3	705.8	100.8	70.6	58.8
	14	62	141.7	681.7	823.4	117.6	82.3	68.6
	16	62	162.0	779.0	941.0	134.4	94.1	78.4
60	10	67	104.7	526.2	630.9	90.1	63.1	52.6
	12	67	125.7	631.4	757.1	108.2	75.7	63.1
	14	67	146.6	736.7	883.3	126.2	88.3	73.6
	16	67	167.6	841.9	1009.5	144.2	101.0	84.1
64	10	76	111.7	596.9	708.6	101.2	70.9	59.0
	12	76	134.0	716.3	850.3	121.5	85.0	70.9
	14	76	156.4	835.7	992.1	141.7	99.2	82.7
	16	76	178.7	955.0	1133.7	162.0	113.4	94.5
66	10	85	115.2	667.6	782.8	111.8	78.3	65.2
	12	85	138.2	801.1	939.3	134.2	93.9	78.3
	14	85	161.3	934.6	1095.9	156.6	109.6	91.3
	16	85	184.3	1068.2	1252.5	178.9	125.3	104.4
70	10	102	122.2	801.1	923.3	131.9	92.3	76.9
	12	102	146.6	961.3	1107.9	158.3	110.8	92.3
	14	102	171.0	1121.5	1292.5	184.6	129.3	107.7
	16	102	195.5	1281.8	1477.3	211.0	147.7	123.1
72	10	102	125.7	801.1	926.8	132.4	92.7	77.2
	12	102	150.8	961.3	1112.1	158.9	111.2	92.7
	14	102	175.9	1121.5	1297.4	185.3	129.7	108.1
	16	102	201.1	1281.8	1482.9	211.8	148.3	123.6

NOTE.—To explain what is meant by a Central space, and to show the arrangement of tubes, reference is made to Figs. 36 and 37, which illustrate clearly the two modes employed.

Three-Inch Tubular Boilers.

TUBES ARRANGED WITH A CENTRAL SPACE.

BOILER.			SQUARE FEET OF HEATING SURFACE.			HORSE POWER. BASED UPON		
Diam. Inches.	Length. Feet.	No. of Tubes.	Shell.	Tubes.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
40	10	28	69.8	219.9	289.7	41.4	29.0	24.1
	12	28	83.8	263.9	347.7	49.7	34.8	29.0
	14	28	97.7	307.9	405.6	57.9	40.6	33.8
	16	28	111.7	351.8	463.5	66.2	46.4	38.6
48	10	44	83.8	345.6	429.4	61.3	42.9	35.8
	12	44	100.5	414.7	515.2	73.6	51.5	42.9
	14	44	117.3	483.8	601.1	85.9	60.1	50.1
	16	44	134.0	553.0	687.0	98.1	68.7	57.2
54	10	54	94.2	424.1	518.3	74.0	51.8	43.2
	12	54	113.1	508.9	622.0	88.9	62.2	51.8
	14	54	132.0	593.7	725.7	103.7	72.6	60.5
	16	54	150.8	678.6	829.4	118.5	82.9	69.1
56	10	54	97.7	424.1	521.8	74.5	52.2	43.5
	12	54	117.3	508.9	626.2	89.5	62.6	52.2
	14	54	136.8	593.7	730.5	104.4	73.1	60.9
	16	54	156.4	678.6	835.0	119.3	83.5	69.6
60	10	68	104.7	534.1	638.8	91.3	63.9	53.2
	12	68	125.7	640.9	766.6	109.5	76.7	63.9
	14	68	146.6	747.7	894.3	127.8	89.4	74.5
	16	68	167.6	854.6	1022.2	146.0	102.2	85.2
64	10	76	111.7	596.9	708.6	101.2	70.9	59.0
	12	76	134.0	716.3	850.3	121.5	85.0	70.9
	14	76	156.4	835.7	992.1	141.7	99.2	82.7
	16	76	178.7	955.0	1133.7	162.0	113.4	94.5
66	10	80	115.2	628.3	743.5	106.2	74.4	62.0
	12	80	138.2	754.0	892.2	127.5	89.2	74.3
	14	80	161.3	879.6	1040.9	148.7	104.1	86.7
	16	80	184.3	1005.3	1189.6	169.9	119.0	99.1
72	10	102	125.7	801.1	926.8	132.4	92.7	77.2
	12	102	150.8	961.3	1112.1	158.9	111.2	92.7
	14	102	175.9	1121.5	1297.4	185.3	129.7	108.1
	16	102	201.1	1281.8	1482.9	211.8	148.3	123.6

NOTE.—To explain what is meant by a Central space, and to show the arrangement of tubes, reference is made to Figs. 36 and 37, which illustrate clearly the two modes employed.

Three and One-half-Inch Tubular Boilers.

TUBES ARRANGED WITHOUT A CENTRAL SPACE.

BOILER.			SQUARE FEET OF HEATING SURFACE.			HORSE POWER. BASED UPON		
Diam. Inches.	Length Feet.	No. of Tubes.	Shell.	Tubes.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
36	10	16	62.8	146.6	209.4	29.9	20.9	17.5
	12	16	75.4	175.9	251.3	35.9	25.1	20.9
	14	16	88.0	205.2	293.2	41.9	29.3	24.4
	16	16	100.5	234.5	335.0	47.9	33.5	27.9
40	10	19	69.8	174.1	243.9	34.8	24.4	20.3
	12	19	83.8	208.9	292.7	41.8	29.3	24.4
	14	19	97.7	243.7	341.4	48.8	34.1	28.4
	16	19	111.7	278.6	390.3	55.8	39.0	32.5
44	10	24	76.8	219.9	296.7	42.4	29.7	24.7
	12	24	92.2	263.9	356.1	50.8	35.6	29.7
	14	24	107.5	307.9	415.4	59.3	41.5	34.6
	16	24	122.9	351.8	474.7	67.8	47.5	39.6
46	10	27	80.3	247.4	327.7	46.8	32.8	27.3
	12	27	96.3	296.9	393.2	56.2	39.3	32.8
	14	27	112.4	346.4	458.8	65.5	45.9	38.2
	16	27	128.5	395.8	524.3	74.9	52.4	43.7
48	10	34	83.8	311.5	395.3	56.5	39.5	32.9
	12	34	100.5	373.8	474.3	67.8	47.4	39.5
	14	34	117.3	436.1	553.4	79.1	55.3	46.1
	16	34	134.0	498.4	632.4	90.3	63.2	52.7
52	10	38	90.8	348.2	439.0	62.7	43.9	36.6
	12	38	108.9	417.8	526.7	75.2	52.7	43.9
	14	38	117.1	487.5	614.6	87.8	61.5	51.2
	16	38	145.2	557.1	702.3	100.3	70.2	58.5
56	10	46	97.7	421.5	519.2	74.2	51.9	43.3
	12	46	127.3	505.8	623.1	89.0	62.3	51.9
	14	46	136.8	590.1	726.9	103.8	72.7	60.6
	16	46	156.4	674.4	830.8	118.7	83.1	69.2
58	10	48	101.2	439.8	541.0	77.3	54.1	45.1
	12	48	121.5	527.8	649.3	92.8	64.9	54.1
	14	48	141.7	615.7	757.4	108.2	75.7	63.1
	16	48	162.0	703.7	865.7	123.7	86.6	72.1
60	10	50	104.7	458.1	562.8	80.4	56.3	46.9
	12	50	125.7	549.7	675.4	96.5	67.5	56.3
	14	50	146.6	641.3	787.9	112.6	78.8	65.7
	16	50	167.6	733.0	900.6	128.7	90.1	75.0
62	10	58	108.2	531.4	639.6	91.4	64.0	53.3
	12	58	129.9	637.7	767.6	109.7	76.8	64.0
	14	58	151.5	744.0	895.5	127.9	89.6	74.6
	16	58	173.1	850.2	1023.3	146.2	102.3	85.3

THREE AND ONE-HALF-INCH TUBULAR BOILERS.—Continued.

TUBES ARRANGED WITHOUT A CENTRAL SPACE.

BOILER.			SQUARE FEET OF HEATING SURFACE.			HORSE POWER. BASED UPON		
Diam. Inches.	Length. Feet.	No. of Tubes.	Shell.	Tubes.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
64	10	60	111.7	549.7	661.4	94.5	66.1	55.1
	12	60	134.0	659.6	793.6	113.4	79.4	66.1
	14	60	156.4	769.6	926.0	132.3	92.6	77.2
	16	60	178.7	879.5	1058.2	151.2	105.8	88.2
66	10	65	115.2	595.5	710.7	101.5	71.1	59.2
	12	65	138.2	714.6	852.8	121.8	85.3	71.1
	14	65	161.3	833.7	995.0	142.1	99.5	82.9
	16	65	184.3	952.8	1137.1	162.4	113.7	94.8
68	10	67	118.7	613.9	732.6	104.7	73.3	61.0
	12	67	142.4	736.7	879.1	125.6	87.9	73.3
	14	67	166.2	859.5	1025.7	146.5	102.6	85.5
	16	C7	189.9	982.2	1172.1	167.4	117.2	97.7
72	10	80	125.7	733.0	858.7	122.7	85.9	71.6
	12	80	150.8	879.6	1030.4	147.2	103.0	85.9
	14	80	175.9	1026.2	1202.1	171.7	120.2	100.2
	16	80	201.1	1172.8	1373.9	196.3	137.4	114.5

NOTE.—To explain what is meant by a Central space, and to show the arrangement of tubes, reference is made to Figs. 36 and 37, which illustrate clearly the two modes employed.

Three and One-half-Inch Tubular Boilers.

Tubes Arranged with a Central Space.

Boiler			Square Feet of Heating Surface.			Horse Power. based upon		
Diam. Inches.	Length. Feet.	No. of Tubes.	Shell.	Tubes.	Total.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
36	10	14	62.8	128.3	191.1	27.3	19.1	15.9
	12	14	75.4	154.0	229.4	32.8	22.9	19.1
	14	14	88.0	179.6	267.6	38.2	26.8	22.3
	16	14	100.5	205.3	305.8	43.7	30.6	25.5
44	10	26	76.8	238.2	315.0	45.0	31.5	26.2
	12	26	92.2	285.8	378.0	54.0	37.8	31.5
	14	26	107.5	333.5	441.0	63.0	44.1	36.7
	16	26	122.9	381.1	504.0	72.0	50.4	42.0
54	10	42	94.2	384.8	479.0	68.4	47.9	39.9
	12	42	113.1	461.8	574.9	82.1	57.5	47.9
	14	42	132.0	538.7	670.7	95.8	67.1	55.9
	16	42	150.8	615.7	766.5	109.5	76.7	63.9
60	10	48	104.7	439.8	544.5	77.8	54.5	45.4
	12	48	125.7	527.8	653.5	93.4	65.4	54.5
	14	48	146.6	615.7	762.3	108.9	76.2	63.5
	16	48	167.6	703.7	871.3	124.5	87.1	72.6
64	10	60	111.7	549.7	661.4	94.5	66.1	55.1
	12	60	134.0	659.6	793.6	113.4	79.4	66.1
	14	60	156.4	769.6	926.0	132.3	92.6	77.2
	16	60	178.7	879.5	1058.2	151.2	105.8	88.2
66	10	62	115.2	568.0	683.2	97.6	68.3	56.9
	12	62	138.2	681.6	819.8	117.1	82.0	68.3
	14	62	161.3	795.2	956.5	136.6	95.7	79.7
	16	62	184.3	908.8	1093.1	156.2	109.3	91.1
72	10	76	125.7	693.3	819.0	117.0	81.9	68.2
	12	76	150.8	835.6	986.4	140.9	98.6	82.2
	14	76	175.9	974.8	1150.7	164.4	115.1	95.9
	16	76	201.1	1114.1	1315.2	187.9	131.5	109.6

Note.—To explain what is meant by a Central space, and to show the arrangements of Tubes, reference is made to Figs. 36 and 37, which exhibit clearly the two modes employed.

Four-Inch Tubular Boilers.

TUBES ARRANGED WITHOUT A CENTRAL SPACE.

BOILER.			SQUARE FEET OF HEATING SURFACE.			HORSE POWER. BASED UPON		
Diam. Inches.	Length. Feet.	No. of Tubes.	Shell.	Tubes.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
42	10	19	73.3	198.9	272.2	38.9	27.2	22.7
	12	19	88.0	238.7	326.7	46.7	32.7	27.2
	14	19	102.6	278.5	381.1	54.4	38.1	31.8
	16	19	117.3	318.2	435.5	62.2	43.6	36.3
48	10	26	83.8	272.2	356.0	50.9	35.6	29.7
	12	26	100.5	326.6	427.1	61.0	42.7	35.6
	14	26	117.3	381.1	498.4	71.2	49.8	41.5
	16	26	134.0	435.5	569.5	81.4	57.0	47.5
52	10	27	90.8	282.7	373.5	53.4	37.4	31.1
	12	27	108.9	339.2	448.1	64.0	44.8	37.3
	14	27	127.1	395.8	522.9	74.7	52.3	43.6
	16	27	145.2	452.3	597.5	85.4	59.8	49.8
54	10	31	94.2	324.6	418.8	59.8	41.9	34.9
	12	31	113.1	389.5	502.6	71.8	50.3	41.9
	14	31	132.0	454.4	586.4	83.8	58.6	48.9
	16	31	150.8	519.4	670.2	95.7	67.0	55.9
60	10	42	104.7	439.8	544.5	79.2	54.5	45.3
	12	42	125.7	527.8	653.5	93.4	65.4	54.5
	14	42	146.6	615.7	762.3	108.9	76.2	63.5
	16	42	167.6	703.7	871.3	124.5	87.1	72.6
64	10	50	111.7	523.6	635.3	90.8	63.5	52.9
	12	50	134.0	628.3	762.3	108.9	76.2	63.5
	14	50	156.4	733.0	889.4	127.1	88.9	74.1
	16	50	178.7	837.8	1016.5	145.2	101.7	84.7
68	10	55	118.7	575.9	694.6	99.2	69.5	57.9
	12	55	142.4	691.1	833.5	119.1	83.4	69.5
	14	55	166.2	806.3	972.5	138.9	97.3	81.0
	16	55	189.9	921.4	1111.3	158.9	111.1	92.6
72	10	60	125.7	628.3	754.0	107.7	75.4	62.8
	12	60	150.8	754.0	904.8	129.3	90.5	75.4
	14	60	175.9	879.6	1055.5	150.8	105.6	88.0
	16	60	201.1	1005.3	1206.4	172.3	120.6	100.5

NOTE.—To explain what is meant by a Central space, and to show the arrangements of Tubes, reference is made to Figs. 36 and 37, which exhibit clearly the two modes employed.

Four-Incn Tubular Boilers.

TUBES ARRANGED WITH A CENTRAL SPACE.

BOILER.			SQUARE FEET OF HEATING SURFACE.			HORSE POWER. BASED UPON		
Diam. Inches.	Length Feet.	No. of Tubes.	Shell.	Tubes.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
40	10	16	69.8	167.5	237.3	33.9	23.7	19.8
	12	16	83.8	201.0	284.8	40.7	28.5	23.7
	14	16	97.7	234.5	332.2	47.5	33.2	27.7
	16	16	111.7	268.0	379.7	54.2	38.0	31.6
50	10	26	87.3	272.2	359.5	51.4	36.0	30.0
	12	26	104.7	326.6	431.3	61.6	43.1	35.9
	14	26	122.2	381.1	503.3	71.9	50.3	41.9
	16	26	139.6	435.5	575.1	82.2	57.5	47.9
62	10	42	108.2	439.8	548.0	78.3	54.8	45.7
	12	42	129.9	527.8	657.7	94.0	65.8	54.8
	14	42	151.5	615.7	767.2	109.6	76.7	63.9
	16	42	173.1	703.7	876.8	125.3	87.7	73.1
72	10	60	125.7	628.3	754.0	107.7	75.4	62.8
	12	60	150.8	754.0	904.8	129.3	90.5	75.4
	14	60	175.9	879.6	1055.5	150.8	105.6	88.0
	16	60	201.1	1005.3	1206.4	172.3	120.6	100.5

NOTE.—To explain what is meant by a Central space, and to show the arrangement of tubes, reference is made to Figs. 36 and 37, which exhibit clearly the two modes employed.

Six-Inch Tubular Boilers.

BOILER.			SQUARE FEET OF HEATING SURFACE.			HORSE POWER BASED UPON		
Diam. Inches.	Length. Feet.	No. of Tubes.	Shell.	Flues.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
40	12	7	83.8	131.9	215.7	30.8	21.6	18.0
	14	7	97.7	153.9	251.6	35.9	25.2	21.0
	16	7	111.7	175.9	287.6	41.1	28.8	24.0
	18	7	125.7	197.9	323.6	46.2	32.4	27.0
	20	7	139.6	219.9	359.5	51.4	36.0	30.0
	22	7	153.6	242.0	395.6	56.5	39.6	33.0
	24	7	167.5	264.0	431.5	61.6	43.2	36.0
42	12	8	88.0	150.8	238.8	34.1	23.9	19.9
	14	8	102.6	175.9	278.5	39.8	27.9	23.2
	16	8	117.3	201.1	318.4	45.5	31.8	26.5
	18	8	131.9	226.2	358.1	51.2	35.8	29.8
	20	8	146.6	251.3	397.9	56.8	39.8	33.2
	22	8	161.3	276.5	437.8	62.5	43.8	36.5
	24	8	175.9	301.6	477.5	68.2	47.8	39.8
46	12	9	96.3	169.6	265.9	38.0	26.6	22.16
	14	9	112.4	197.9	310.3	44.3	31.0	25.9
	16	9	128.5	226.2	354.7	50.7	35.5	29.6
	18	9	144.5	254.5	399.0	57.0	39.9	33.3
	20	9	160.6	282.7	443.3	63.3	44.3	36.9
	22	9	176.6	311.0	487.6	69.7	48.8	40.6
	24	9	192.7	339.3	532.0	76.0	53.2	44.3
48	12	11	100.5	207.3	307.8	44.0	30.8	25.7
	14	11	117.3	241.9	359.2	51.3	35.9	29.9
	16	11	134.0	276.5	410.5	58.6	41.1	34.2
	18	11	150.8	311.0	461.8	66.0	46.2	38.5
	20	11	167.6	345.6	513.2	73.3	51.3	42.8
	22	11	184.3	380.1	564.4	80.6	56.4	47.0
	24	11	201.1	414.7	615.8	88.0	61.6	51.3
50	12	12	104.7	226.2	330.9	47.3	33.1	27.6
	14	12	122.2	263.9	386.1	55.2	38.6	32.2
	16	12	139.6	301.6	441.2	63.0	44.1	36.8
	18	12	157.1	339.3	496.4	70.9	49.6	41.4
	20	12	174.5	377.0	551.5	78.8	55.2	46.0
	22	12	192.0	414.7	606.7	86.7	60.7	50.6
	24	12	209.4	452.4	661.8	94.5	66.2	55.2

SIX-INCH TUBULAR BOILERS—Continued.

BOILER.			SQUARE FEET OF HEATING SURFACE.			HORSE POWER BASED UPON		
Diam. Inches.	Length. Feet.	No. of Tubes.	Shell.	Flues.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
54	12	14	113.1	263.9	377.0	53.9	37.7	31.4
	14	14	132.0	307.9	439.9	62.8	44.0	36.7
	16	14	150.8	351.9	502.7	71.8	50.3	41.9
	18	14	169.7	395.8	565.5	80.8	56.6	47.1
	20	14	188.5	439.8	628.3	89.8	62.8	52.4
	22	14	207.4	483.8	691.2	98.7	69.1	57.6
	24	14	226.2	527.8	754.0	107.7	75.4	62.8
60	12	15	125.7	282.7	408.4	58.3	40.8	34.0
	14	15	146.6	329.9	476.5	68.1	47.7	39.7
	16	15	167.6	377.0	544.6	77.8	54.5	45.4
	18	15	188.5	424.1	612.6	87.5	61.3	57.1
	20	15	209.4	471.2	680.6	97.2	68.1	51.7
	22	15	230.4	518.4	748.8	107.0	74.9	62.4
	24	15	251.3	565.5	816.8	116.7	81.7	68.1

FIVE-FLUE BOILERS.

BOILER.		Diam. of Flues.	SQUARE FEET OF HEATING SURFACE.			HORSE POWER BASED UPON		
Diam. Inches.	Length. Feet.	Inches.	Shell.	Flues.	TOTAL.	7 sq. Feet.	10 sq. Feet.	12 sq. Feet.
36	12	7	75.4	110.0	185.4	26.5	18.5	15.5
	14	7	88.0	128.3	216.3	30.9	21.6	18.8
	16	7	100.5	146.6	247.1	35.3	24.7	20.6
38	12	7	79.6	110.0	189.6	27.1	19.0	15.8
	14	7	92.8	128.3	221.1	31.6	22.1	18.4
	16	7	106.1	156.6	252.7	36.1	25.3	21.1
	18	7	119.4	164.9	284.3	40.6	28.4	23.7
40	12	8	83.8	125.7	209.5	29.9	21.0	17.5
	14	8	97.7	146.6	244.3	34.9	24.4	20.4
	16	8	111.7	167.5	279.2	39.9	27.9	23.3
	18	8	125.7	188.5	314.2	44.9	31.4	26.2
	20	8	139.6	209.4	349.0	49.9	34.9	29.1
42	14	9	102.6	164.9	267.5	38.2	26.8	22.3
	16	9	117.3	188.5	305.8	43.7	30.6	25.5
	18	9	131.9	212.1	344.0	49.1	34.4	28.7
	20	9	146.6	235.6	382.2	54.6	38.2	31.9
	22	9	161.3	259.2	420.5	60.1	42.1	35.0
44	14	10	107.5	183.3	290.8	41.5	29.1	24.2
	16	10	122.9	209.4	332.3	47.5	33.2	27.7
	18	10	138.2	235.6	373.8	53.4	37.4	31.2
	20	10	153.6	261.8	415.4	59.3	41.5	34.6
	22	10	169.0	288.0	457.0	65.3	45.7	38.1
	24	10	184.3	314.2	498.5	71.2	49.9	41.5
46	16	10	128.5	209.4	337.9	48.3	33.8	28.2
	18	10	144.5	235.6	380.1	54.3	38.0	31.7
	20	10	160.6	261.8	422.4	60.3	42.2	35.2
	22	10	176.6	288.0	464.6	66.4	46.5	38.7
	24	10	192.7	314.2	506.9	72.4	50.7	42.2
48	16	10	134.0	209.4	343.4	49.1	34.3	28.6
	18	10	150.8	235.6	386.4	55.2	38.6	32.2
	20	10	167.6	261.8	429.4	61.3	42.9	35.8
	22	10	184.3	288.0	472.3	67.5	47.2	39.4
	24	10	201.1	314.2	515.3	73.6	51.5	42.9

TESTIMONIALS.

THE TOLEDO BRUSH ELECTRIC LIGHT AND POWER CO.

TOLEDO, OHIO, FEB. 14th, 1883.

The Cummer Engine Co., Cleveland, O.

GENTS:—In answer to your question as to how we are satisfied with our engine, we answer as follows:

After a thorough investigation of engines, we purchased a Cummer Engine, because we believed it to be more closely governed than any other engine we could find, as we desired it for the use of the electric arc lights, and the steadiest possible power is desirable in the use of these lights, in order to produce steady lights; for any slight variation of speed, however small, will instantly be seen in the lights; hence it is absolutely necessary that the engine be self-governing, and governed accurately.

In the use of our engine we are constantly turning off and on lights. Frequently we turn on or off a dynamo electric machine running forty lights and requiring about thirty-five horse-power. This is sometimes done by a friction pulley, and sometimes by simply closing a current with a switch. If done in the latter way, the full weight of the dynamo comes upon the engine instantly. This forty light dynamo machine requires about thirty-five horse-power and instantly throwing on or throwing off this thirty-five horse power machine does not affect the motion of our engine sufficiently to make it perceptible in the lights.

We expected much from this engine from what we had heard of it before purchasing, but it exceeds all our expectations, and is entirely satisfactory.

It is also very economical in the use of fuel, and is not liable to get out of order.

We have had our engine about a year, and have never had occasion to stop it for a moment in consequence of any thing getting out of order.

This is a very important point in our business, for if the engine stops a minute, all the lights are extinguished for a minute.

For steadiness, for economy, and for absence of liability to get out of order, we believe the Cummer Engine is unequalled.

Respectfully yours,

J. W. POST,

Sec'y and Treas. Toledo Brush Electric Light and Power Co.

[The above refers to one of our 14x30 Standard Engines rated by us at 100 H. P. using 90 pounds steam cutting offat $\frac{1}{5}$ stroke.

CUMMER ENGINE CO.]

MR. W. C. STOEPEL, Sec'y and Treas. THE MICHIGAN MALLEABLE IRON CO., Detroit, Mich., says:

"Your 10x30 Automatic Cut-off Engine which we purchased of you has given us excellent satisfaction, and we can speak of it in terms of the highest praise. We required, as we supposed, about 40 horse power for our ultimate work, expecting to run fourteen rolling barrels and one fan. We are already runnin g *twenty-five* barrels and *two* fans, and are at times using in the neighborhood of eighty horse power. Still it plods on easily and with great regularity, and performs its extra work with apparent ease. It works with great economy of steam and governs very closely under sudden variations in the load—as for instance, when starting or stopping the fans or rolling barrels. The engine is simple of construction; its valves are very easily moved, and are readily exposed. Since first starting up, last spring, it has been run every working day and some nights, and as yet we have been to no expense for repairs."

OFFICE OF MOORE & SONS, MERCHANT MILLERS.

BUNKER HILL, KAN., FEB. 12th, 1883.

F. D. Cummer, Esq., Cleveland, O.

DEAR SIR:—The engine we purchased of you one year ago, we find upon thoroughly testing it, to be the most economical both in fuel and water we have ever seen, and for giving steady power and regularity of motion it has no equal, the Governor especially performing its work to perfection. We are yours, &c.,

MOORE & SONS.

BOSTON, Jan'y 29th, 1883.

Cummer Engine Co., Cleveland, O.

GENTLEMEN:—In reply to your inquiry as to our reasons for adopting the Cummer Engine in our business, we would say that we have had a large experience in Automatic engines not only in this country but abroad. Our Mr. Hill was the Massachusetts Commissioner to the Vienna and to the Philadelphia Expositions, and a careful student at the late Paris Exposition, and we are wholly familiar with all the improved cut-off engines, of any importance, built in Europe as well as in this country.

We have a large business in engines of this class, our sales for the last three or four years have run into several hundredthousand dollars, and our business is largely with the large cotton mills and other corporations requiring from 100 to 1000 H. P., and which are the most

particular customers in the country both as to economy and construction of their engines.

Having heard good reports of the character of your engine, we made careful examinations of engines which you have running and decidedly made up our minds that you had the most sensible, practical and most satisfactory valve gear that we have ever seen.

We then went through your new shops at Cleveland and satisfied ourselves, that with the admirable tools which you have there collected or built, and with your thorough and practical experience, we might rely upon first class design and workmanship in these engines, and we were the more pleased, because you decided to prepare and build these engines to suit the most modern ideas now in vogue in England and in the East.

Having given the matter careful consideration, we could not but decide to identify ourselves with you and to arrange to represent you in the East, and we feel very sure that we have the very engine with which to compete with the high class engines here built, in perfect regulation, in economy and above all in that exact workmanship which is the first requisite for a successful engine in New England.

Since we came to this conclusion, we have been much gratified to learn that Gen. Leggett, late Commissioner of Patents, holds opinions so coincident with ours that he has invested largely and has permanently identified himself with your Company. Hoping this will answer your inquiry, we remain Yours truly,

(Signed) HILL, CLARKE & CO.

RICH HILL, MO., MARCH 3d, 1883.

NORDYKE & MARMON CO., INDIANAPOLIS, IND.

The Cummer Automatic engine which you sold us together with the 150 barrel roller mill outfit is the best made, and will give more power with less fuel than any other engine having the same size of cylinder.

As for motion we think it is as near perfection as can be, and is not liable to get out of order any more than an ordinary slide valve engine. Every one who notices it says it does its duty well.

Yours,

ELIAS TALOR & SONS.

NOTE.—This engine is a 10x30 furnished N. & M. Co. and forwarded by them to destination. The starting was done by Messrs Elias Talor & Sons without any assistance on our part.

CUMMER ENGINE CO.

HATCH & MITCHELL, GRAND RAPIDS, MICH., say in regard to a 12x30 engine of our manufacture:—

"We are very poor judges of engines as this is the only one we have ever used and, therefore, cannot speak intelligently in regard to its details of construction, but in regard to regularity of motion we don't see how it can be bettered as there is no perceptible difference in motion with steam anywhere from 40 to 80 lbs.

The expenses have been light thus far; from what we learn in regard to other engines in this city we are inclined to believe the Cummer is equal to any if not superior."

INDIANAPOLIS, IND., Jan. 18th, 1883.

F. D. Cummer, Esq.,
CLEVELAND, OHIO.

DEAR SIR:—We have, within the last few months, started up several large new roller process flouring mills which were built by us, and in some we are using your Automatic Engines to drive the machinery. Feeling certain that you would be pleased to hear your machinery praised, (we have the same weakness,) we wish to say that we are very much pleased with the engines, finding them very sensitive under extreme changes in loads, very simple in construction, hence not liable to get out of order. We find your smaller sizes peculiarly adapted to small sized flouring mills in the Western States, where fuel is extremely high.

Very truly yours,

NORDYKE & MARMON CO.,

Manufacturers of Flouring Mill Machinery.

MR. J. C. THORNTON, engineer at the Model Mill, GRAND RAPIDS, MICH., says:—

I am glad to speak a word in favor of the Cummer Engine. I consider it to be a grand success and second to none now in the market, for easy running and perfect governing. I never saw an engine whose governor controlled the speed so perfectly at different changes of load as the "Cummer Engine." It effects a great saving in steam, and for simplicity, close governing and economy it stands the test for all that is claimed for it. I cheerfully recommend your engine in preference to any engine now in use.

J. C. THORNTON,

Engineer Model Mills.

GRAND RAPIDS, MICH.

From a personal letter from Mr. J. L. Booth, engineer of the Brush Electric Light and Power Company, of Montgomery, Ala., we make the following quotation :

" As engineer of the Brush Light and Power Company, of Montgomery, Ala., I feel that I cannot say too much in reference to your engine and its action. Its governing is faultless under any change of load, while its economy is so satisfactory that we have taken no pains to make an actual test. I am more than satisfied with the results.

Respectfully yours

J. L. BOOTH.

The following letter was written in reply to an inquiry from Minnesota as to the performance of the Cummer Engine :

" We are using a Cummer Condensing Engine ; have used it for two years. It is 14x30 and is doing from 100 to 130 H. P. of work. We run night and day, and only stop it 10 or 15 minutes out of 24 hours. We have never laid out over $5 worth of repairs on it in the two years. Any common engineer can run one and can safely say it is the closest governing engine and the most economical engine built anywhere. There is never any trouble about its running away. In fact, the governor is the most perfect one built ; and for anyone that wants to purchase a first-class, durable and economical engine, we cheerfully recommend them to purchase a Cummer Automatic Cut-off Engine. We think they are far ahead of the Corliss."

Yours Resp'y,

Signed. MORRISON MF'G CO., St. Joseph, Mich.

D. M. Morrison, V. P. & Secy.

P. S.—To anybody that has to buy fuel: We saved the cost of this engine in fuel in 2 years. We had a common engine before.

AMES & HUMPHREY, MILLERS, RUSSELL, KAN., say :

" Your 14x30 Automatic Engine which was put in our mill last January, is giving us good satisfaction. It runs very smooth and almost noiseless. The governor can't be beat for giving a steady motion. We can see no change when our four runs are on or off, or when the steam varies from 25 to 30 lbs. We are saving more than one-third of the coal used by our old engine."

The following letter is a portion only, of an unsolicited letter from the owners of the engine.

J. C. BEACH & BRO.

WALKILL, N. Y.

General Office, 240 Broadway, N. Y.

The engine was started up last Saturday and does our work with ease, much to our satisfaction.

Mr. Eckliff, has taken off several diagrams with the indicator, some which he has no doubt sent you. Mr. Eckliff expresses himself well pleased with the working, and we can say so far as we are able to judge, the engine is all you claim for it. The movement certainly is uniform, and the amount of work seems to make no difference with the number of revolutions.

At the Borden Condensary in this place they have a Corliss engine. Their engineer thinks ours much the better. Our own engineers, who have been accustomed to steam engines and have had the care and working of them, say they never saw an engine use so little steam. Shall hope to hear from the diagrams soon. Yours truly,

(Signed.) J. C. BEACH & BRO.

A few months afterward we wrote Messrs. J. C. Beach & Bro., to which they replied as follows:

WALLKILL, N. Y., Jan. 12th, 1883.

F. D. Cummer, V. P., &c.

DEAR SIR:—Your favor of Dec. 27th was received and mislaid, hence the delay in reply.

We have to say that the engine put in our mill and started up some time in July (aside from the little trouble with the cross head gibs), is entirely satisfactory, it works quietly, smoothly and is the best governed, and as easily handled as any engine we ever had to do with. We take pleasure in showing it to any one when running, and cheerfully recommend it to any one about to put in steam power. Were we to put in another engine would certainly give yours the preference.

Yours truly,

J. C. BEACH & BRO.

MR. GEO. M. HOAG, Sup't of the Evansville, Ind., Brush Electric Light and Power Co., says:—

"It gives me great pleasure to say that your engines here (a pair of 14 x 30 engines coupled and so arranged that one or both engines may be used), are working to our entire satisfaction."

OFFICE OF UNION WADDING CO.,
PAWTUCKET, R. I., June 18, 1882.

Mr. F. D. Cummer.

DEAR SIR:—In reply to your inquiry as to the working of the steam engine you sent us, a 14x30, we would say that it fully comes up to the high expectations we had formed of it before purchasing. It runs smoothly, governs admirably under all conditions, and whilst we cannot say absolutely as to its economy, as it is supplied by the same set of boilers as supply several other engines, we are perfectly satisfied from the exhaust and cards taken from it that it is very economical to use. The extreme simplicity of its construction and the ease with which it operates the valves, commend it strongly to us, and we feel safe in recommending it to any one as one of the best of first-class engines.

We understand that you are about to enlarge your works for the contruction of them, and we wish you all success, as we believe in the "Cummer Engine."

Most respectfully yours,

UNION WADDING CO.

H. A. Stearns, Supt.

The Union Wadding Company, at the time our engine was purchased by them, were familiar with all the eastern automatic engines, through living as they do amongst the many works and factories that make and use them. They besides have had in use several styles of the best, and were and are now using a 400 H. P. Geo. H. Corliss Engine.

CUMMER ENGINE CO.

THE MICHICAN STOVE COMPANY, of Detroit, Mich., says: "Our 20x36 Cummer Condensing Engine, with a load that varies at times from 50 per cent. to 100 per cent., has, by actual count, a variation of speed of only one and a half revolutions. The saving in fuel is over 40 per cent. There are no stops or delays on account of the engine. It has run every day for two years, except Sundays, holidays, and stock taking. The engine is a success in every way; we could not be better pleased, and if we ever want a new engine, our preference is for the 'Cummer.'"

[NOTE.—The great variations of load referred to is in part caused by the turning on and off of the power necessary to operate a very large Root Blower, which takes 67 H. P., as shown by the indicator.

CUMMER ENGINE CO.]

GRAND RAPIDS, Mich., Dec. 30, 1882.

F. D. Cummer, Esq., Cleveland, O.

DEAR SIR:—Yours of 27th inst. received. In reply would say that our Engine is working satisfactory. It runs very smooth and does not appear to make any effort in running. We cheerfully recommend it to any one that may want a first-class engine.

Yours truly,

GRAND RAPIDS PLASTER CO.

WM. S. HOVEY, Agent.

LOCKLAND, O., Sept. 16th, 1882.

Cummer Engine Co.

GENTLEMEN:—As to working of our 14 x 30 "Cummer Engine," which we have had in use about four months, would say it has all along and is still doing good service, working economically and entirely to our satisfaction in every way. It governs closely, runs smoothly, causes no stop or delay, and we think it will keep up its record right along.

Respectfully,

(Signed.) LOCKLAND LUMBER CO.

The engine referred to above displaced a leading automatic engine, as the extreme variations of load required closer governing than the displaced engine could give.

WM. HARRISON, HARRISON WAGON WORKS, Grand Rapids, Mich., says:

"My 18x36 Cummer Condensing Engine runs very satisfactorily. Is always on hand. Variations of load very great, but governor gives a very uniform and satisfactory motion. Is so economical that one man cares for the engine and does the firing, and has more leisure than any man around the establishment."

IRON CLAD PAINT COMPANY.
75 & 77 Central Way,
CLEVELAND, O., Nov. 28, 1882.

Robert Suppiger, Esq.
HIGHLAND, Ill.

DEAR SIR:—In reply to yours of 26th inst., asking information as to what induced our firm to adopt the Cummer Engine and information as to how it is performing, I would say: In the first place we had contracted for an engine, boiler, etc., with another party, which proved so poor that we rejected it and compelled him to remove it. We were then in

need of an engine and went inquiring for a good one. We had heard of the Cummer and made inquiries as to such engines, and found it highly recommended. We were acquainted with Mr. R. N. Allen, the inventor of the Allen Paper Car Wheel; had known him for more than thirty years; knew him when he was M. M. on the old Cleveland, Norwalk & Toledo R. R. (now part of the L. S. & M. S. R'y.) and knew him to be a good engineer and mechanic, so we inquired of Mr. Allen for the best engine for us to buy. He recommended the Cummer Engine and recommended us to have Mr. Cummer take the job and set the engine to running. Mr. Allen told us that his Co. (the Allen Paper Wheel Co.) had purchased three engines of the Cummer make and that they were all first-class engines and gave perfect satisfaction. He further said that he knew Mr. Cummer to be a first-class mechanic, and that he would do as he agreed; that the engines were economical, were well built and not apt to get out of order; that they were smooth running and in fact the best engines to buy. Mr. Allen told us that he had no interest in the Cummer Engine Co. nor any other Engine Co., and he simply recommended it because he believed it would give satisfaction and was the best engine for us to buy. These recommendations and others induced us to adopt the Cummer Engine, and we are perfectly satisfied and only regret that we did not get it in operation early last spring instead of being delayed trying another kind which proved bad.

The engine is working entirely satisfactory. Our foreman thinks it is the best engine he ever saw. If we were in need of another engine we would buy another of the same kind. I will request Mr. Stevens to write you, stating how it is performing as he can tell better than I can.

Very truly yours,

JAMES WADE, *Treasurer, &c.*

CLEVELAND, Nov. 29, 1882.

Robert Suppiger, Esq.

DEAR SIR:—Mr. Wade, our treasurer, has answered explaining reasons for adopting the Cummer Engine. As to its performance I will say that it has given us the very best of satisfaction, both for regularity of speed and economy. I think it is one of the best governed engines in the market. We are perfectly satisfied, and if we had occasion to put in a second engine it would be a Cummer.

Very Respectfully,

C. M. STEVENS, *Foreman.*

Messrs. R. SUPPIGER & Co. purchased from us an 18x36 Standard Engine for their Highland Mill.

CUMMER ENGINE CO.

DETROIT, MICH., May 8th, 1883.

The Cummer Engine Co., Cleveland, Ohio.

GENTLEMEN: The 16x36 engine furnished us last November has been set up, and is now running, and we are pleased to say, that so far as tested, it comes up to our expectations in all respects. The engine presents a handsome appearance; the workmanship is excellent throughout, and its performance seemingly justifies all that you claim, or that favorable reports from other localities had led us to look for. It performs its work easily, and runs smoothly and quietly—the governor, especially, acting with remarkable promptness and certainty, keeping the engine up to its standard number of revolutions, better than any other governor with which we are acquainted. We cannot speak with certainty about economy as yet, because no actual test has been made; but when the amount of coal consumed is compared with the work which we do, we feel hopeful that it will prove a very economical engine. We can express ourselves as being well pleased with our purchase, and if nothing happens to change our present views, should wish to give you the preference when we need more power.

Yours Respectfully,

CLOUGH & WARREN ORGAN CO.

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